The
ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

VOL. XVIII No. 7 JULY, 1957

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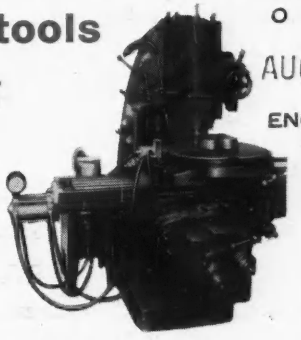
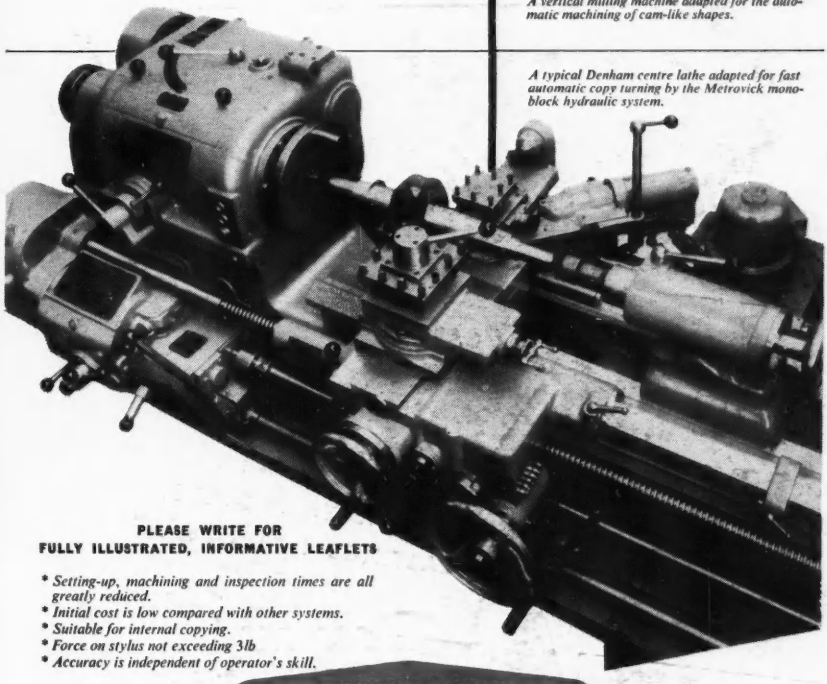


A vertical milling machine adapted for the automatic machining of cam-like shapes.



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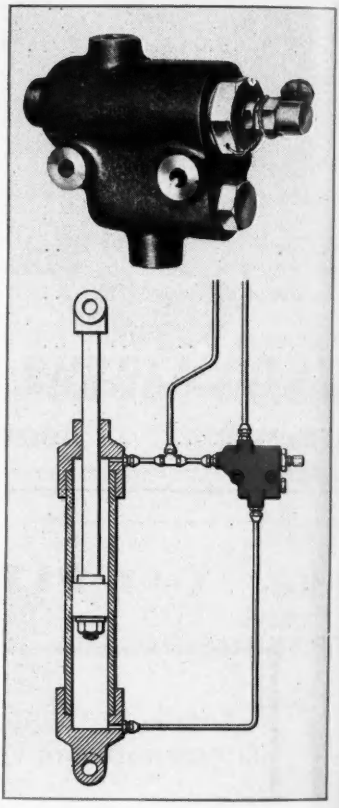

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THE BLUE PAGES

A QUICK GLANCE AT THE LATEST TECHNICAL DEVELOPMENTS THROUGHOUT THE WORLD

● *HIGH-VELOCITY FORMING TECHNIQUES*

For the forming of especially tough metals such as certain fully hard stainless steels, special high-velocity methods are now being developed, some of which are based on the use of detonating charges. The various approaches to high-velocity metal working can be classified under three main headings, i.e., (1) the use of shaped charges which form the metal with half a die or none at all; (2) gas-discharging cartridges which force a piston or punch through the metal; and (3) expansion of high-pressure gases which drive opposed rams. A high-velocity impact press working on the last-named principle has been built and is pneumatically operated with dry nitrogen at 1200 psi. Two cylinders simultaneously force two steel heads towards each other, and the impact forces produced are said to exceed 10^6 in.-lb, the closing velocity being in excess of 200 fps. A die is mounted on one head and a rubber pad on the other, the impact force being completely absorbed by the part being formed. A similar design has been developed for punching holes through metals with tensile strengths in the 300,000-psi range. It is claimed that the holes produced are superior to those drilled, and consideration is now being given to the design of a machine for punching rivet holes. In order to prevent stress cracks around the hole, the punch may be so shaped as to coin a depression around each hole. The general consensus of opinion of experts working on problems of high-speed forming appears to be that the forming presses eventually developed will be much smaller than present types, and it is expected that not only will tools last ten to twenty times longer than those used in conventional methods, but that powder- or cartridge-actuated tools will replace a substantial portion of the portable tools now in use.

● *THREE-DIMENSIONAL PRINTED-WIRING TECHNIQUE*

The employment of ferrite cores makes it possible to construct exceedingly large memory arrays for computer devices. Thus, with the employment of a $256 \times 256 \times 38$ stack of ferrite cores, no less than $2\frac{1}{2}$ million binary digits can be stored. It will be readily appreciated that the manual wiring of such assemblies presents a formidable task, and hence the importance of the development of three-dimensional printed wiring based on photo-etching techniques cannot be exaggerated. The new technique is based on the use of collimated light sources with which, in a single exposure, a latent image of the complete wiring pattern for a memory plane can be produced. The advantage of the use of collimated light, i.e., a beam of parallel light rays, rests on the fact that the mask and the photo-sensitive surface can be separated, while with ordinary light the negative mask must be in intimate contact with the surface. The image can thus be projected with fidelity onto any surface, regardless of its irregularity, for effecting exposure of the photo-sensitive surface. Before exposure the entire memory-plane assembly must be coated with copper, which is deposited on a plastic coating used to anchor the ferrite cores in place. The initial layer of copper adheres firmly to the plastic as a highly conductive base, on which a thicker layer of copper is then electro-deposited, and an acid-resist is next applied to the entire assembly. Photographic exposure of the sensitized surface through the masks is made with four collimated light sources, which form contact images of the surface wiring on both sides of the assembly, as well as projection images of the lines connecting the surfaces through each ferrite core. A concept of the precision required in carrying out the process may be gained from the fact that each ferrite core through which an etched wiring connection is to be made is 0·05 in. i.d., the cores being located at 0·1-in. centres.

● *PLATING STEEL TUBES WITH MOLYBDENUM*

A recent report describes a processing equipment developed for plating the inner wall of steel tubing with molybdenum, a plating thickness of 0·004 in. having been achieved. In a typical test run, the internal surfaces of the tubes were bored to $1\frac{5}{8}$ in. and finished to better than 16 micro-inches r.m.s. After careful degreasing with acetone, the tubes were given thin underplatings of copper, nickel, or nickel and cobalt, chemical underplatings of cobalt proving the best. During the molybdenum plating process, the tubes were heated, the external temperature in the hottest zone ranging between 875° and 975°C. For most of the

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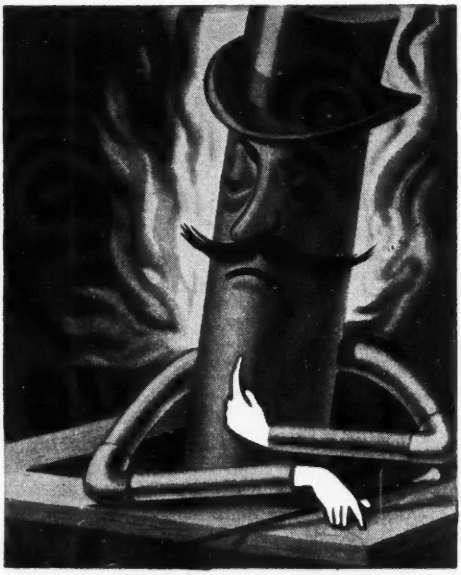

"I am a four-inch bore pipe. I work in a factory. It is my job to carry steam at 450°F. from generator to the plant. But I was a waster, a fuel waster, a *very* fuellish waster indeed!

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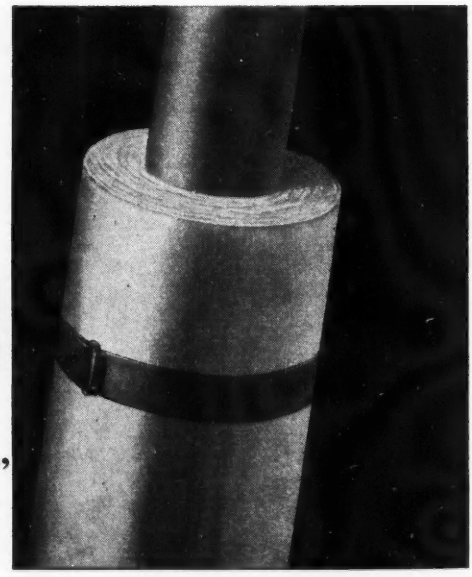

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experiments an annular high-temperature gas burner proved satisfactory. Mounted on a moving platform, so that it could be passed over the length of the tube, the burner produced a narrow band of intense heat which could be controlled to close limits. Infra-red lamps were used on the unheated portions of the tube, so as to prevent condensation of the molybdenum pentachloride used as the plating medium. The molybdenum pentachloride was derived from a vaporizer in which argon as the vaporizing gas was bubbled through the molten pentachloride, the maximum operating temperature being 250°C. A carefully controlled amount of hydrogen was added to the vapour prior to its admission to the tube to be plated. After admission, the reaction of the hydrogen with the molybdenum pentachloride vapour in the hottest zone produces the molybdenum plating. The ratio of the molybdenum pentachloride to the hydrogen appears to be one of the fundamental variables in the process, as low vapour concentrations were found to increase plating time. Lean mixtures produce dense, well-bonded platings, while a high ratio of pentachloride to hydrogen leaves the coating porous and non-adherent, although much thicker.

● *MEASUREMENT OF PRESSURE DISTRIBUTION ON ROTATING BLADES OF TURBO-MACHINES*

Measurements on rotating blades of turbo-machines are of considerable interest, because the effect of the rotation of the row and the consequent centrifugal forces produced on the flow of the working medium have so far been explored only slightly. It would therefore appear that the results of a more thorough investigation such as that conducted recently constitute an important contribution to the three-dimensional theory of axial-flow turbo-machines. This investigation was made with a special test rig which enabled a survey of the pressure distribution over the surface of the blades of rotating blade rows to be made, the rotor blades being made of a fusible plastic and glued into an aluminium ring. Each blade was provided with twenty cast-in pressure conduits, and connections to each of these conduits were drilled normal to the surface of the blade, so that twenty holes were distributed over the front and back of the profile in each surface of measurement. Altogether, seven surfaces of measurement were provided at various radii from tip to root of the blade, i.e., 140 holes per blade. The pressure distribution in only one of these surfaces was measured in any one test run, the pressure holes of the remaining six being closed off with thin adhesive tape. The twenty pressure leads were connected to terminals on a disc rotating with the wheel and then transmitted by a selector mechanism to a disc rotatable relative to the wheel. The latter disc was operated by an electric motor mounted inside the turbine rotor. The connecting sliding surfaces of the selector mechanism were designed to prevent any leakage, enabling the pressures to be transmitted without error from the rotating turbine shaft through a rotating pressure seal. From the latter, pressures were successively supplied to one of twenty manometers by a distributor and selector disc of similar design to that in the turbine.

● *HIGH-TEMPERATURE LIGHTWEIGHT MATERIAL FROM GLASS*

A recent announcement gives a brief description of a new non-porous crystalline material made from glass which may prove to be a strong competitor for metal as a basic engineering material, as it is claimed to be harder than high-carbon steel, lighter than aluminium, and up to nine times stronger than plate glass. It will retain its strength at temperatures up to 1300°F, its softening point being given as 2460°F, which is above the melting point of some stainless steels. In addition, the material is said to have good electrical insulating properties and resistance to acid corrosion. Furthermore, it is claimed to have great flexibility in forming, since it can be pressed, drawn, blown, rolled, and centrifugally cast. Commencement of commercial production is imminent and, while the material will be more expensive than glass, it will be less expensive than stainless steel. In view of the great interest which attaches to the development of such an extraordinary material, release of additional information will be looked forward to with more than ordinary interest.

● *HIGH-TEMPERATURE ARC PLASMA*

In a report in our issue of July 1952, attention was drawn to the development of high temperature water-stabilized arcs. In this type of arc the arc column is controlled or stabilized by a surrounding water vortex confined in a vortex chamber, the arc plasma leaving the vortex chamber through a centrally located nozzle. According to a report on the latest research into this type of arc, the highest verified temperature measured thus far has been 14,000°K, and this temperature has been brought to bear on objects by directing upon them the plasma jet, which has a speed of some 4000 fps. This plasma flows from the arc in a highly dissociated state and can be described as an atomic vapour, since complete molecular breakdown occurs. With such a plasma jet, steady-state heat-transfer rates as high as 2000 BTU/sq ft/sec, at a distance of some four nozzle diameters from the nozzle, have been reached, as compared with 800 BTU/sq ft/sec for rocket-engine nozzles. It is considered that, with such

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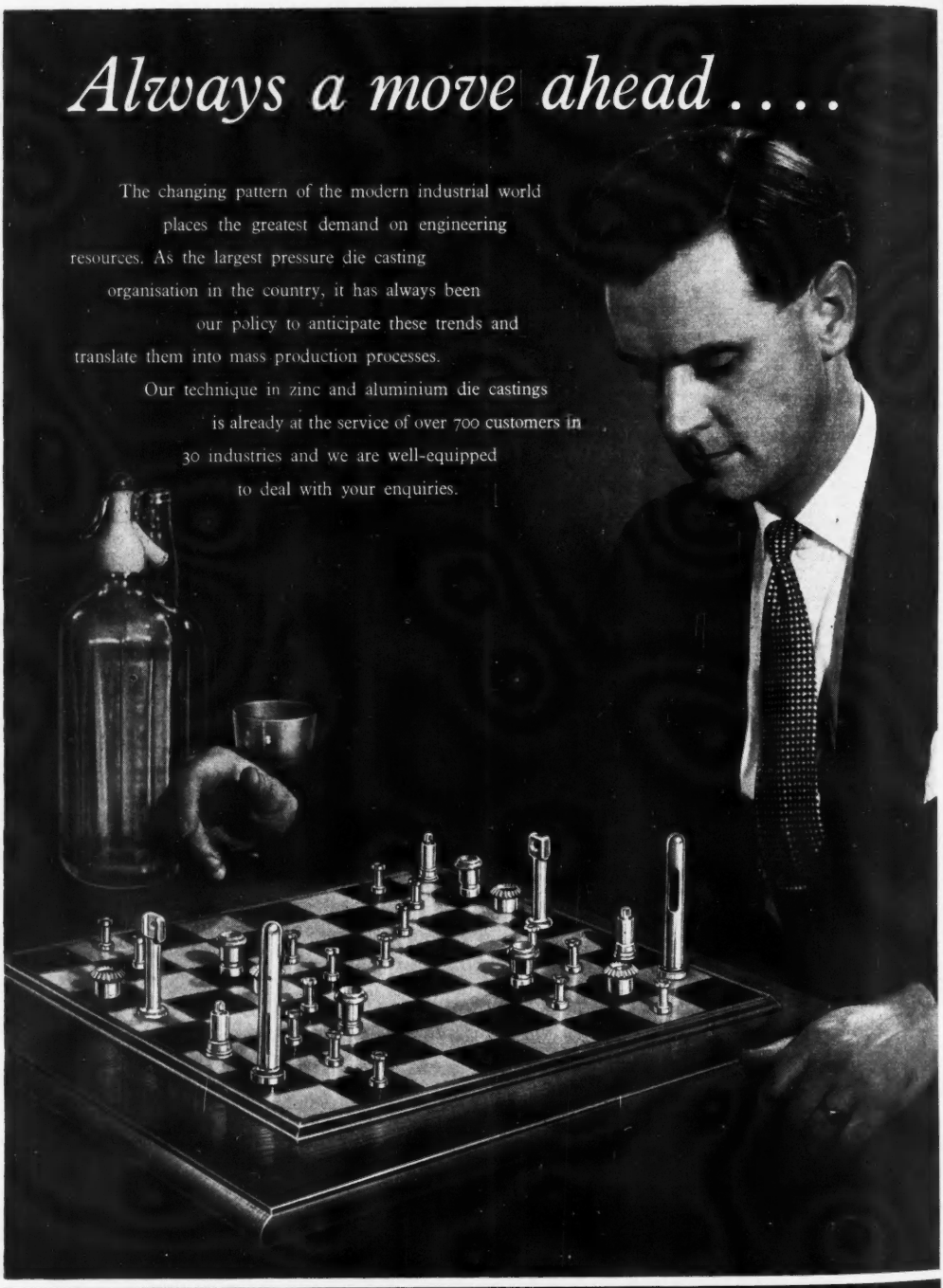

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high temperatures, velocities, and heat-transfer rates, the arc constitutes a very useful means of testing materials which may be exposed to equivalent conditions in service. The report gives additional information on a recently developed air-vortex-stabilized arc with which velocities as high as 10,000 fps can be achieved. Tests with stabilized arcs of either type should prove superior to rocket-engine tests, which, besides giving much lower temperatures, are almost prohibitively expensive in routine testing. While the expense of a stabilized arc is very low, its power requirements are very high. For a ¼-in. arc some 80 kW are required, while a 1¼-in. arc requires 3000 kW, and a 3-in. arc about 8000 kW. The importance of the development is indicated by the fact that construction of equipment for producing an 8000-kW arc is actually being planned.

● ***METHOD OF MAKING IMPROVED JOINTS BETWEEN ELECTRICAL CONDUCTORS***

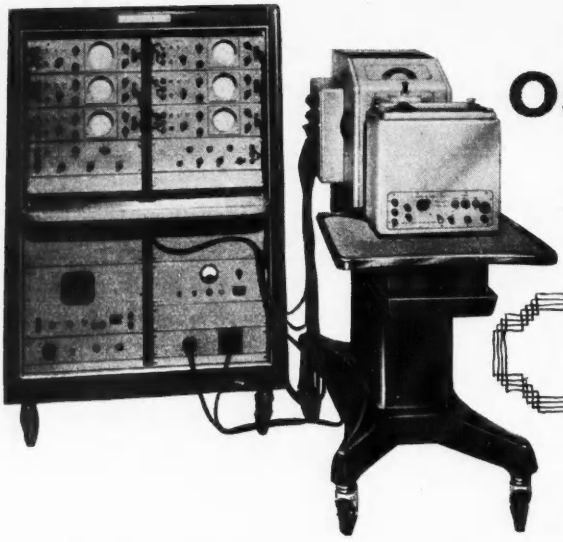
A recent patent specification describes an improved method of making mechanical joints between electrical conductors such as aluminium bus-bars. It is known that the formation of efficient joints between conductors of oxidizing metals requires the removal of the oxide skin, which has a high electrical resistivity, and the exclusion of air from the deoxidized mating surfaces while the joint is being made. The formation of a natural oxide skin is particularly marked in the case of aluminium and aluminium-base alloys, and various methods have hitherto been used to overcome its effects on the efficiency of conductor joints. Such methods include scratch-brushing under vaseline, tinning with solders, electroplating, and metal spraying. It is claimed that none of these methods is entirely satisfactory from the point of view of electrical conductivity, stability during service, cost, or ease of application. The proposed new method comprises the application to one or both of the mating surfaces of a jointing composition containing a substance which will effect the removal of the oxide skin, and also a salt or oxide of a metal which will become deposited on the surface as the result of electrochemical displacement. This jointing composition remains permanently between the mating surfaces of the joint. In order to effect deposition of the metal on the joint surface by electrochemical displacement, the metal incorporated in the jointing composition is so chosen that it is situated lower in the electrochemical series than the conductor metal. Preferably, a wetting agent is incorporated in the composition, and a dyestuff or pigment may also be included, so as to indicate clearly the area of metal which has been treated. As an example, using a composition containing 0·7% zinc oxide, 3·5% sodium hydroxide, 5·7% water, 90% calsolene oil, and 0·1% colouring matter, the electrical resistance of a joint in an aluminium bus-bar was 0·8 unit, assuming the resistance of a plain bus-bar to be 1 unit, while that of the untreated joint was 5·0 units.

● ***STRETCH-ORIENTED TRANSPARENT PLASTICS***

It has been found that multi-axial stretching of acrylic plastic sheeting and certain other plastic materials greatly increases their resistance to stress crazing and to stress-solvent crazing. In fact, this resistance increases with increasing degrees of stretching, and it has been observed that the increase in crazing resistance is accompanied by a large increase in the strain at failure and a slight increase in tensile strength. Little effect on the tensile modulus of elasticity has been noted, but some decrease in resistance to surface abrasion has been found. Annealing increases the tensile strength of the stretched materials slightly, and resistance to stress-solvent crazing of both stretched and unstretched materials is usually increased markedly by annealing. There appears to be a close relationship between crazing and molecular orientation ; thus, crazing may be assumed to start at submicroscopic flaws or weak points in which by chance the polymer chain segments are oriented normal to the applied tensile stress. With sufficient stress, a separation between portions of adjacent chains occurs, and the resulting crack grows until it reaches a region in which the polymer chain segments are oriented approximately in the direction of the tensile stress. Subsequent crack growth may involve rupture of primary valence bonds, especially if the stress is relatively high, of the order of the tensile strength. In multi-axial stretching, the chain segments apparently turn into a position more nearly parallel to the surface. As the orientation increases, it may become more difficult for a submicroscopic crack to propagate through the thickness of the sheet, because of the development of cleavage planes.

● ***HARDENABILITY TEST FOR DEEP-HARDENING STEELS***

Steels used in the manufacture of very large sections must possess deep-hardening properties. In testing the hardenability of such steels, the standard Jominy test specimen 1 in. in diameter by 4 in. in length, hardens along practically its full length and gives no information concerning the limiting hardenability of the steel. The simplest type of test would be to use an oversized Jominy specimen, 3 in. in diameter by 12 in. in length, but, in end-quenching such a specimen, it has been found that air cooling is predominant at positions greater



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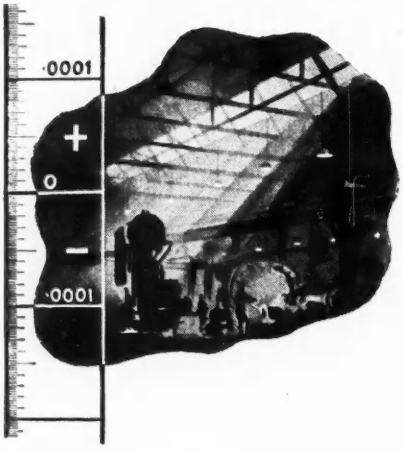

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than 2 in. from the quenched end. In a newly developed test procedure the specimen is a bar, 1½ in. in diameter by 11 in. in length, having seven thermocouple wells along its length and a supporting stem. It is placed in a three-zone vertical tube furnace, and power to each zone is controlled independently by manually operated variacs. Two small end heaters and an equalizer tube of stainless steel, in conjunction with the independent power control in each zone of the furnace, permit a uniform temperature to be maintained along the specimen for austenitizing. In conducting the test the specimen is heated to the proper austenitizing temperature and soaked for 1 hr. Power to the end heaters is then cut off and the bottom heater is withdrawn. The specimen is lowered 3 in. through the furnace and is end-quenched with a standard Jominy stream. The equalizer tube must also be lowered to surround the specimen during quenching, except for from ⅛ to ¼ in. at the bottom end. In the meantime, the furnace is cooled by a programme controller in a cycle which corresponds to the water-quenching effect at the 3-in. position of a 12-in. Jominy specimen.

● *CASTING TECHNIQUE USING PROGRESSIVE FREEZING*

It is reported that castings with the physical properties of forgings are now possible through the use of anodized aluminium moulds with a special pouring technique which produces progressive freezing from the bottom up. The new method is said to be applicable to all usual casting alloys based on steel, cast iron, magnesium, and aluminium. The process differs from sand casting in that a metal mould is used, and from permanent-mould casting in that both metal feed rate and cooling pattern are controlled. The results are claimed to be improved and uniform properties throughout the casting, elimination of such internal defects as porosity and shrinkage, and freedom from surface and sub-surface defects. Reduced machining time and good surface finish are claimed as added advantages. There are no inherent limitations to the size of casting which can be made by the new process. The largest casting yet made on a production basis is of an aluminium alloy ; it is over 5 ft in diameter and weighs 350 lb, but even larger castings are reported to have been produced successfully on a development basis. On the other hand, castings weighing a fraction of an ounce are claimed to have been produced with equal success by this process, which makes use of the high thermal conductivity of aluminium to control both the direction and rate of heat transfer from the molten metal. The mould is made of aluminium, and the face which comes into direct contact with the molten metal is deeply anodized. The anodized layer has a melting point of over 3700° F and is also hard and therefore resistant to erosion and wear, making it possible to use the mould repeatedly. Being a refractory material with low thermal conductivity, the anodized film momentarily checks the flow of heat from the metal being poured, keeping it from freezing too soon. Thus, the metal which is first poured into the mould is the first to freeze, so that controlled pouring produces progressive solidification of the part from the bottom up, the last to freeze being the riser.

● *IMPROVEMENT OF PRINTING BY ULTRASONICALLY VIBRATED TYPE-BARS*

A recent patent specification refers to the observation that low-velocity typewriter pressure printing is much less effective in producing carbon copies than high-velocity impact printing, even with identical peak pressures in both cases. Presumably, the shock waves resulting from the high-velocity impact aid in transferring the carbon vehicle in a stack to the next succeeding page to be printed. On the other hand, the noise level from impact printing is much higher than that of pressure printing, and accordingly impact printing is not the complete solution to multiple-copy work. The patent in question aims at providing an improved method of printing which combines the advantages of low noise level associated with low-pressure printing with the improved carbon-copy production achieved with impact-type printing. Briefly, the invention consists of a new printing method in which the type-bar is first brought into the printing position and then pressed with moderate pressure against the sheet, after which it is vibrated at a very high frequency. This is carried out by providing a linkage mechanism in which an ultrasonic transformer is one component. It is claimed that, with this equipment, ultrasonic vibrations of only a few milliseconds duration make it possible to produce good carbon copies with a minimum noise level. Moreover, the high-frequency vibration transferred to the type-face creates heat in the paper stack and tends to melt the wax vehicle used in carbon paper, so that the carbon is transferred substantially in a liquid state to succeeding pages in the stack. It is stated that the heat generated by the ultrasonic vibrations is also sufficient to permit printing on a heat-sensitive paper. The application of the new method is suggested not only with regard to typewriters but also to multi-copy printing presses.

For further information on developments reviewed in THE BLUE PAGES, readers are invited to write to the Editor.

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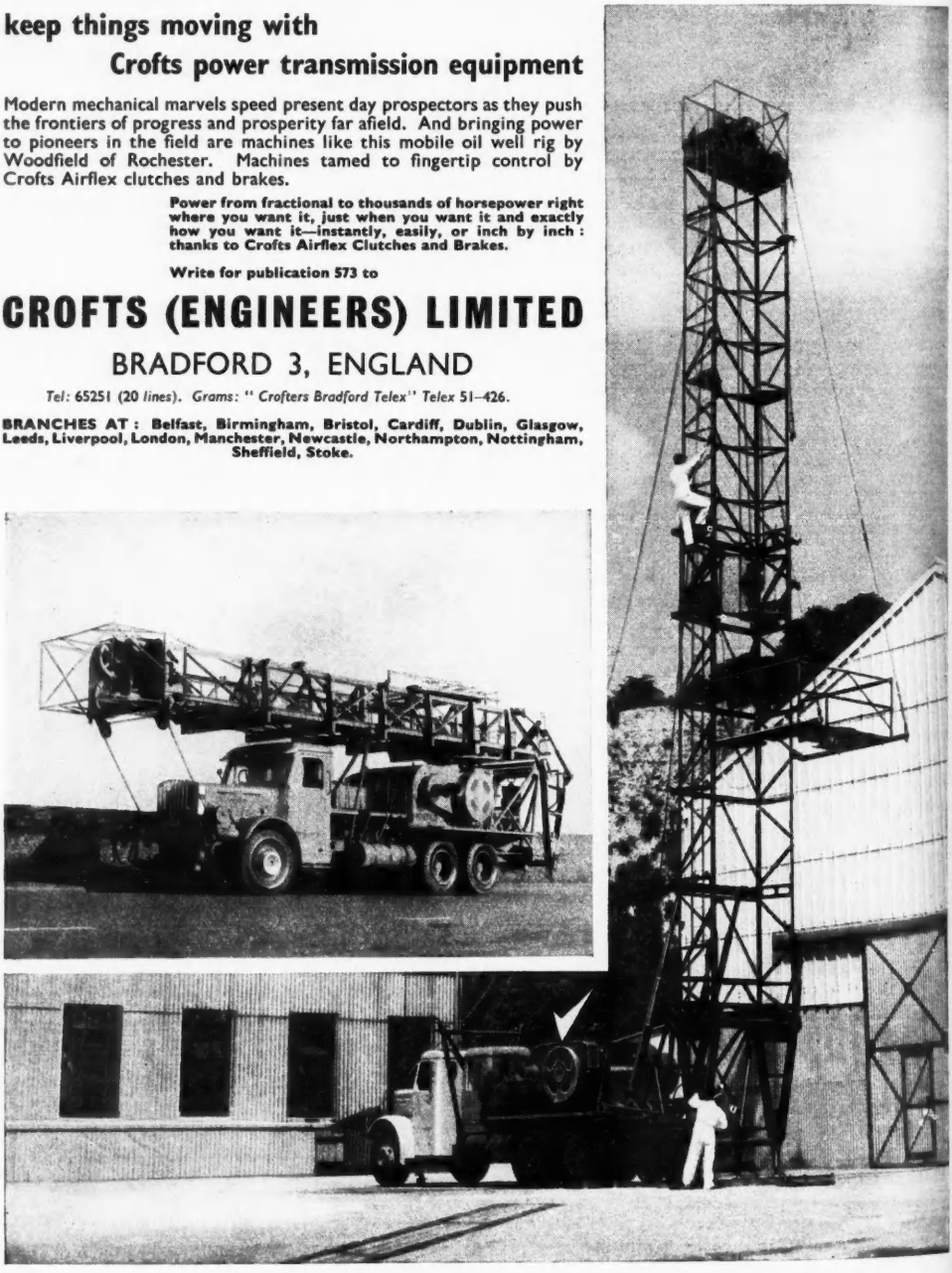

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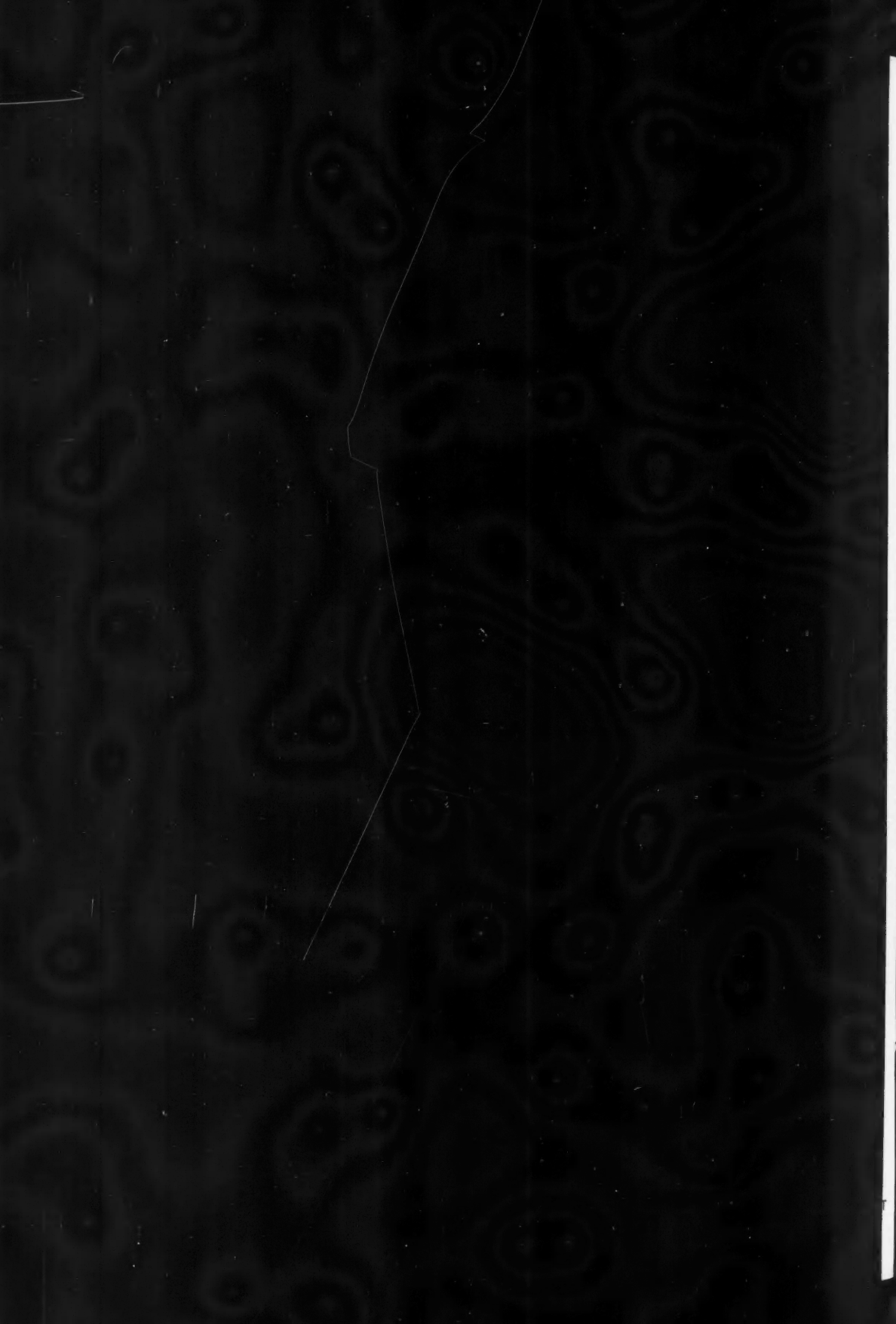

ENGINEERS' DIGEST SURVEY No. 2

Flexible Couplings

An Editorial Staff Report

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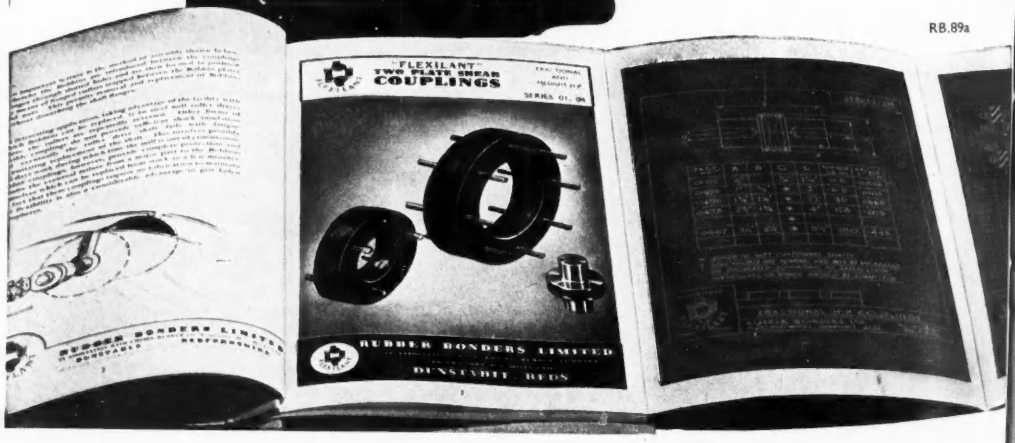


Flexible Couplings

AN EDITORIAL STAFF REPORT

THE purpose of this survey is to provide users of flexible couplings with descriptions of the various types now commercially available, to indicate their main features, and to set forth some basic criteria for the selection of couplings according to particular applications.

The subject is limited, in principle, to flexible couplings giving a permanent positive drive; consequently, it does not include clutches of any kind, which, for the purpose of this survey, are defined as couplings which can deliberately be engaged and disengaged whilst in motion ; also, it does not deal with universal joints, which will be considered separately in a subsequent survey, although couplings allowing for appreciable angular misalignment are basically equivalent to these in some respects.

Specific Properties of Flexible Couplings

Rigid coupling arrangements, e.g., bolted, keyed, press-fitted, or shrunk-on connections, can be employed without too much difficulty for comparatively short lengths of large-diameter shafts running in bearings with brackets or pedestals on a common bedplate. In these cases, extreme care is always required to ensure proper lining up, so as to avoid dangerous cyclic bending stresses in the shaft. On composite beds or subdivided foundations, however, perfect alignment becomes much more difficult to achieve and maintain, owing to eventual wear and gradual movement and distortion of supports, etc. In all such systems, therefore, it is an advantage to use non-rigid coupling arrangements. With flexible couplings, a certain amount of misalignment is permissible, because the forces due to angular, radial (or parallel), or axial errors (Fig. 1) are either made inoperative by joints or links, or by pivoting or sliding action, or give rise to strains which are associated with low (i.e., acceptable) stresses in the flexible members of the coupling.

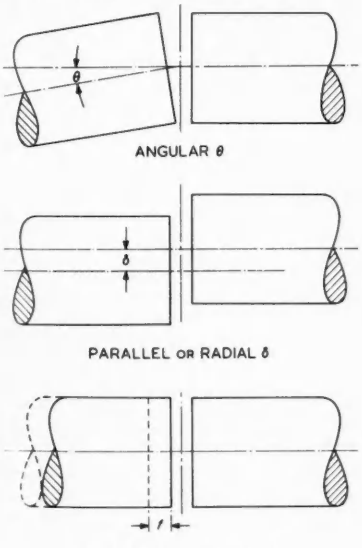


Fig. 1. Diagrammatic representation of angular, parallel, and axial misalignments.

Although the majority of flexible couplings were originally designed merely to cope with certain amounts of misalignment, it is now recognized that they can also be called upon to fulfil various other requirements, such as smoother running conditions and reduction of the effects of impact, vibration, and intermittent sudden changes of load. In this connection, these requirements may lead to considerations of the vibratory behaviour and transient response of mechanical systems. Whatever virtues flexible couplings may have, apart from the correction of misalignments, there are cases where they can be wholly effective only if a system is properly studied for vibratory conditions at the design stage. Consequently, optimum results with couplings are obtained only when such systems are analysed in conjunction with coupling manufacturers, who are always ready to give the benefit of their experience.

A list of the specific properties, features, and characteristics which, to some extent, all couplings are designed to possess is given below:—

(A) Torque-carrying capacity.
(B) Accommodation of axial, radial (or parallel), and angular misalignment.
(C) Reduction of vibration, impact loads, and the effects of sudden intermittent load changes.
(D) Reduction of backlash effects.
(E) Protection against overload torques.
(F) Insensitivity to atmospheric or other operating conditions.
(G) Durability, wear resistance, etc.
(H) Comparative ease of assembly and disassembly.
(I) Operation in either direction of rotation.

This list is not, of course, exhaustive, since it is difficult to assess all possibilities in the wide range of designs so far developed ; moreover, some requirements may conflict with others, so that couplings, like other components, are usually a compromise between requirements of which the relative importance varies according to the particular application.

(A) Torque-Carrying Capacity

In makers' catalogues, the torque-carrying capacity is generally given in pound-inches of torque T at various specified values of shaft speed N in rpm. Alternatively the horsepower P at each speed is given. As is well known, the equation relating these quantities is

$$T = 63{,}000\ P/N$$

In all cases, the maximum permissible running speed is also indicated. Some makers give different horsepower ratings for their couplings, according to whether they are to be used under " normal service " conditions (e.g., for centrifugal compressors or pumps, generators, propellers, conveyors, and machine tools), or "severe service " conditions (e.g., for reciprocating compressors and pumps, crushing machinery, sawmills, and textile spinning frames). Frequently, the safety factor recommended in such cases is $1\frac{1}{2}$ or 2. This is sufficient in many cases ; however, for couplings subjected to vibrations, rough running conditions, and impact torques, higher safety

factors are necessary, possibly attaining several times these values.

It can be said that the safety factor to be employed depends both on the driving and driven machine, as a smooth-running prime mover may be coupled with a rough-running driven machine or *vice versa*. Hence, both ends of the drive must be considered. If R_1 = safety factor of the prime mover, and R_2 = safety factor of the driven machine, then the overall safety factor S can be taken as $S = R_1 + R_2$. The coupling with a maximum torque rating T will then have an effective torque-carrying capacity

$$T_{\text{eff}} = T/S$$

Purely as a comparative example of the variations which can be encountered, the following is a list of the safety factors for prime movers and driven machines quoted by one manufacturer :—

PRIME MOVERS (R_1)	
Turbines and electric motors	0
Steam and petrol engines	1·3
Paraffin engines	1·5
Gas engines, 2- and 4-cylinder	2·5
Gas engines, 1-cylinder	2·7
Diesel engines, 6-, 7-, and 8-cylinder	2·0
Diesel engines, 4-cylinder	2·25
Diesel engines, 1-, 2-, 3-, and 5-cylinder	3·0
DRIVEN MACHINES (R_2)	
Steelworks Plant:—	
Forging machines (belt- or chain-driven)	2·0
Wire mills	2·0
Shearing machines	3·0
Bar-straightening machines	3·5
Tube mills	3·5
Forging machines (direct-driven)	4·0
Plate mills and finishing mills	3·0–3·5
Bar mills	4·0–4·5
Roughing mills	4·5–5·0
Steelworks Cranes (*Reversing and with Brake Drums*) :—	
Auxiliary and main cross traverses	2·0
Long travel	2·5
Main and auxiliary hoists and slewing	3·5
Grinding, Crushing, and Screening Plant :—	
Rotary screens	1·3
Grinding mills	1·5
Ball mills and coal pulverizers	2·0
Cement mills and disintegrators	2·25
Rubber mixers and cane knives	3·0
Joggling machines	3·5
Ore crushers (large gear reduction)	4·0
Roller cane mills	4·0
Wood- and Metal-Working Machinery :—	
Light wood-working machinery	1·25
Machine tools (excluding planing machines)	1·25
Wood-planing machinery	1·5
Sawing and heavy wood-working machinery	2·0
Metal planing machines (reversing)	2·5
Conveying and Hauling Machinery :—	
Belt conveyors	1·25
Automatic boiler stokers	1·5
Winches and capstans	2·0
Winders and haulages	2·5
Passenger and goods lifts	3·5
Paper-Making Machinery :—	
Rotary screens	1·3
Brush doctors and drying cylinders	1·5
Couch rolls and press rolls	2·0
Calender rolls	2·7
Pulpers	3·5
Pumping and Compressing Plant :—	
Exhausters	1·2
Blowers, fans, and centrifugal pumps	1·25
Rotary compressors	1·5
Radial-type compressors	2·0
Mine-ventilating fans	2·5
Ram pumps and reciprocating compressors	3·0
Miscellaneous :—	
D.C. Generators	1·25
Textile machinery	1·25
Alternators	1·3
Welding generators	2·0
Bar-reeling machinery	2·5
Electric or hydraulic steering gear	2·5
Marine propellers	2·5

The above values, as already stated, are for comparative purposes only, as certain couplings can be associated with higher or lower effective torques, depending on the coupling design and the particular application. In fact, safety factors vary from manufacturer to manufacturer, sometimes to a considerable degree, and it is therefore desirable in all cases to consult the makers and obtain their recommendations.

Finally, it should be noted that some manufacturers prefer to use safety factors for the driving and driven machines, which are to be multiplied, instead of added, to obtain the overall safety factor, so that, in this case, $S = R_1 \times R_2$.

(B) Accommodation of Misalignment

When considering particular types of couplings regarding their ability to cope with misalignments, it is an important point that any misalignment be absorbed by the coupling components themselves, i.e., that no end thrust, side thrust, additional axial or bending vibrations, torsional vibrations, edge loads, or rubbing action be imposed on the shaft. When the ability to accommodate misalignment is limited in order to improve other properties, provision for particularly accurate alignment should be made or, alternatively, for floating or sliding action elsewhere in the shaft system.

For low power ratings, small-size couplings are available which can take up a considerable amount of misalignment, e.g., flexible fabric or rubber disc couplings or sleeve couplings. With increased power, however, the difficulties are somewhat greater.

For the absorption of angular misalignment, bush or pin couplings, rubber-block couplings, gear couplings, and chain couplings are particularly suitable. In large metal or rubber disc couplings, angular or axial misalignment may in some designs result in additional bending distortion of the disc, making it less effective in sustaining torsional loads. Additional capacity to deal with misalignment can be obtained by employing two couplings at either end of an intermediate shaft, this being in effect a cardan arrangement. If the joint angle is greater than, say, ½ deg., the cardan effect may result in cyclic speed fluctuations on the output shaft of a frequency twice that of the running speed. Irrespective of the ability of a coupling to take up misalignments, there-

fore, every care should be taken to install couplings in such a way as to avoid large amounts of misalignment. Indeed, as a general rule, the initial alignment of the shafts to be coupled together should always be as good as possible ; otherwise, the initial plus the additional misalignment which may arise as a result of operational displacements may easily exceed permissible limits.

(C) Reduction of Vibration, Impact, and the Effects of Sudden Intermittent Load Changes

It is worth emphasizing that, except when very low speeds and torques are contemplated, e.g., in couplings between instrument components, tuning controls of radio receivers, stirring devices, and drives for miniature equipment, it is not advisable to interpose a coupling merely on the basis of its flexibility as a means of taking up misalignment. This practice has led to failures not only in the coupling but sometimes also in the machines themselves.

Particularly when the driving torque is non-uniform, or when the size and moment of inertia of the driven machine are comparable with those of the prime mover, the torsional-vibration conditions of the entire system should be investigated before finally deciding on any particular coupling. In this connection, a number of points which are sometimes overlooked are as follows :—

(1) Couplings should be calculated not only to sustain the mean torque, but also the maximum torque which may arise in the system due to gas pressure, inertia, impact, etc. If there is any uncertainty in this respect, a safety factor of, say, 2 to 5, depending on the roughness of the drive, should be used.

(2) Couplings should be firmly connected, e.g., keyed to shafts, when any appreciable power is to be transmitted; Woodruff keys should be avoided, since they cut too deeply into the shaft. Alternatively, splined connections or pressure fits may be used. Grub-screws, set-screws, etc. are mainly suitable for low and steady torques, and can in some cases be used as safety arrangements for overload protection.

(3) The " stiffness " or torsional rigidity of a coupling is defined as the torque required to produce unit angular deflection. A figure for coupling stiffness is, of course, required in torsional-vibration calculations, as well as the values of the moments of inertia of the two main coupling members, except when they are negligible in comparison with other inertias in the system. Coupling makers frequently quote a figure for stiffness based on static tests at low loads. For high loads or torques, it is sometimes suggested that the static stiffness be multiplied by a factor of 1·5 or 2. This uncertainty may cause measured frequencies of vibration and critical speeds to have values different from those expected from the calculations, sometimes with unfortunate results. Certain coupling manufacturers guarantee their stiffness figures within a specified percentage, and users are entitled to ask for such guaranteed figures.

The range of torques within which the stiffness of a coupling can be regarded as practically constant is known as the " linear range ". In this range a straight-line relation exists between torque and deflection. Constant-stiffness or linear couplings are the easiest to assess in calculations, but variable-stiffness couplings are preferred in many cases. Thus, a rubber block in compression has a practically constant stiffness only for moderate loads ; at high loads, its resistance increases appreciably. Hence, for large fluctuating or vibratory torques, it will show a variable stiffness. Similarly, if two springs are placed in series, one being much more flexible than the other, deflections under low loads will occur mainly in the softer spring, and the two-spring arrangement will have a low stiffness. Above a certain load, the soft spring will become coil-bound, and any further deflection will occur in the stronger spring ; at high loads, this arrangement will therefore have a high stiffness.

Now, if such a system is properly calculated for an installation in which there would otherwise be critical speeds at the lower and upper end of the speed range, and if the load increases with running speed, the installation may be tuned in such a way that, with low stiffness at low loads, the low-speed criticals will fall below the minimum running speed, and the high-speed critical, owing to increased coupling stiffness, will be displaced beyond the upper limit of the speed range.

Another important property of couplings with variable stiffness is their " detuning " action, i.e., their ability to alter the frequency of vibration of the entire system during each vibration cycle as the coupling swings from a positive to a negative deflection maximum, passing through its minimum stiffness value at zero deflection. As a result, the coupling will prevent the build-up of resonant conditions in the system, so that torsional-vibration amplitudes and stresses will be kept at acceptably low values.

Variable-stiffness couplings produce remarkably good results, provided that they are employed under suitable conditions. However, the use of any type of coupling for difficult conditions with pulsating loads is a problem, and, in order to benefit from the improvements made possible by variable-stiffness couplings, such problems should preferably be solved by experts or by the coupling manufacturers.

(4) The statement that flexible couplings eliminate vibrations, impacts, and the effects of sudden load changes is, of course, too general to be universally valid, and expected improvements must be substantiated by practical experience in every case.

Reduction of impact loads can be obtained with a soft rubber coupling or with a highly flexible spring coupling, if the inertia of the driven machine is greater than that of the prime mover. If the inertia of the driven machine is much smaller than that of the prime mover, the interposition of a soft coupling will not cause impact loads from the driven end to affect the drive unit, but any sudden load or speed variations produced by the latter will have a strong impact effect on the driven machine. The question as to what should be done in this case is not easily answered. If a spring coupling is used, with a certain amount of slack deliberately introduced into its design, impact may be taken up by the slack in the coupling. If a soft rubber coupling is used, it may have sufficient damping to lessen the impact. Should these properties still be insufficient for a particular application, it may be found necessary to introduce a friction drive or some other arrangement capable of slipping under excessive torque. Another possibility is to include viscous or magnetic damping in the driven parts of the system. Vibration dampers are used successfully not only in engine systems, but also in machine tools.

To sum up, vibrations can be reduced by means of a coupling only if the latter has stiffness, detuning, or damping characteristics which are suitably matched with those of the proposed system. Whereas detuning action is a very powerful means of vibration reduction, coupling damping (in mechanical couplings) is a somewhat uncertain factor and should not necessarily be expected. Impacts can be reduced by couplings of certain types in

some circumstances. Sudden load changes are also difficult to handle ; where frequent load changes are contemplated, it may be desirable to use slip-type couplings to reduce acceleration effects on the driven machines.

D) Reduction of Backlash Effects

Trouble associated with backlash effects occurs in transmission gears, in push-rod and rocker types of valve-timing gears, and in machine-tool mechanisms of various types, when the vibratory inertia forces acting on contact members are greater than the steady transmitted force. Some of the possibilities for overcoming loss of tooth contact, chatter, and noise due to backlash can be outlined as follows:—

(1) It is advisable to ensure, as far as possible, that there will always be sufficient steady force (or torque) transmitted from the driver to avoid force (or torque) reversals. Therefore, idling conditions at low speeds and low transmitted loads require particular attention, although in some systems high-speed conditions may also be troublesome.

(2) As regards torsional or axial vibrations, as well as cyclic speed fluctuations causing chatter, the application of a highly flexible coupling will only be successful if the driven inertia is much greater than that of the driver ; in this case, most fluctuations are absorbed in the coupling. For instance, if a soft coupling is followed by a large flywheel connected to a gearbox, it may be expected that engine-speed fluctuations will not reach the gearbox. In the opposite case, i.e., if a small pinion in contact with a larger gear-wheel is preceded by a soft coupling connected to large prime-mover inertias, any drive fluctuations will be accentuated by the soft coupling and will give unsatisfactory results at low loads and idling speeds.

(3) Although in some cases, in view of the operating characteristics of the system, the possibility of torque reversal cannot be entirely eliminated at the design stage, it may not materialize in practice, if a variable-stiffness coupling (which detunes the inertia torque), a coupling with a high damping capacity, or a spring or chain coupling incorporating a certain amount of slack in its components is used. Another method sometimes adopted is to provide two sets of parallel input gears and shafts which are preloaded in opposite directions, so as to maintain tooth contact with the main gear-wheel. This solution is mainly practicable in drives with inherently flexible shafts.

(4) In particularly difficult cases, it may be necessary to employ a disengaging coupling, i.e., a friction, magnetic, or hydraulic clutch, which will lessen the possibility of backlash effects at idling speeds, at the expense, however, of some loss of power due to slip in parts of the speed range.

In all these cases, there is no general solution, and it is unjustifiable to ascribe all causes of trouble to manufacturers of gears, couplings, or engines, when, in fact, a partial re-design of the system might well be the real remedy.

(E) Protection against Overload Torques

Torques due to sudden load changes and impacts of an intermittent nature have already been discussed. The overload torques considered in the following are those extreme torques which may occur if there is partial seizure or a sudden excessive drop in speed of the prime mover or the driven machine. In such cases, the machine which has not developed the failure requires immediate protection against the sudden torque peak. Usually, this is obtained by a special design feature incorporated into the coupling, which breaks off the mechanical connection between the main units.

As distinct from slip-type couplings, permanent-engagement couplings are employed to avoid any slip, so as to maintain full transmission efficiency, and overload protection is obtained at the cost of a temporary or permanent rupture of some part of the coupling. In couplings incorporating rubber in shear or tension, or comparatively thin metal springs subjected to bending, overload causes stresses high enough to produce failure of the material or of the bond between rubber and metal. For couplings employing rubber in compression (e.g., rubber-block couplings), for bush couplings, for comparatively heavy springs or fully enclosed spring packs, for gear couplings, and for metal disc couplings, shear pins or shearing keys are usually employed. These pins or keys are made of copper, copper alloy, or aluminium, i.e., of metals having a lower shear modulus than that of the materials in which they are seated. Rupture will then occur in these elements, without appreciable damage to the remainder of the coupling or the shaft. Although these elements are calculated to rupture at a specified load, they may, however, fail before this load is reached, since the fatigue strength of materials is a statistical average value which may be exceeded or which may not be reached by individual components. Furthermore, there is the possibility that, owing to repeated application of moderate overloads, the shearing elements will be subjected to fretting, crushing with permanent deformation, pounding, and abrasion after a prolonged period of service. The shearing elements may then fail prematurely. However, a temporary breakdown involving only a localized failure is preferable to coupling destruction, and shearing elements will therefore continue to be recommended, wherever necessary.

Instead of shear pins, a friction-controlled arrangement can be used for overload protection. Basically, this arrangement consists of a non-metallic disc pressed against one of the flanged hub members by a metal disc, the assembly being held together by spring-loaded elements. The permissible amount of overload torque is controlled by the spring pressure. As in the case of shear pins, the critical amount of overload can be determined only approximately.

Another type of overload-protection device, recently incorporated into couplings, makes use of the differential expansion of inner and outer hub members, i.e., push-fit cone rings are loosened as a result of the temperature rise occurring with increased friction due to excessive torque, and thus disconnect the two portions of the coupling.

In certain applications, the installation may incorporate an intermediate unit, e.g., a gearbox, which itself may require overload protection. If the overload is expected to occur on the driven end of the system, the coupling and/or shear-pin arrangement should be provided on the output side of the gearbox. If impulses apt to cause sudden reversals or overloads can also originate at the driving end, the gearbox requires protection at its input side. Thus, in difficult cases, couplings with overload protection may be needed on both sides of the gearbox.

(F) Insensitivity to Atmospheric or other Operating Conditions

Couplings are obliged to operate under a wide variety of conditions, the main factors to be considered being

dust and grit, temperature, exposure to water, oil, or other liquids, and centrifugal forces at high running speeds.

It may be said that, generally, dust and grit do not appreciably affect rubber-type couplings, as they cannot penetrate rubber surfaces. Metal disc couplings are also inherently insensitive to dust. Spring couplings and couplings incorporating linkage arrangements, as well as some other couplings, however, can be fully protected only if they are totally enclosed. Gear couplings require special protection against the ingress of dust, to safeguard against loss of oil film on contact surfaces.

Temperature insensitivity, within certain limits, is a property which can be associated with all-metal flexible couplings. Provided that suitable lubricants are used, gear couplings can operate from sub-zero temperatures to temperatures above 300°F, as is also the case with oil-filled or oil-lubricated metal spring couplings. Some types of fully enclosed synthetic-rubber couplings can also be used at fairly high temperatures. For temperatures between, say, 15 and 175°F, obtaining near an engine under normal working conditions, natural-rubber couplings or rubber fabric couplings are particularly suitable.

Exposure to water does not generally affect rubber couplings, whereas metal couplings usually require adequate covers and seals. Oil and oil-mist do not affect all-metal couplings, but in rubber couplings the rubber parts must be protected from direct contact with oil, for instance, by enclosing them in metal casings or by applying a protective oil-resistant coating to the exposed surfaces. All couplings intended for operation in special liquids or corrosive atmospheres need special sealing and protection.

Centrifugal forces set a limit to the running speeds of certain types of couplings, e.g., rubber sandwich couplings, link couplings, and rubber sleeve couplings. Only comparatively small-diameter couplings can be employed for very high-speed drives, e.g., gas turbines, since hoop stresses are proportional to the square of the running speed. If high power is to be transmitted at such high speeds, gear-type and certain spring-type couplings are usually the most suitable ; for moderate power at high speeds, bonded-rubber or double-sleeve couplings can be considered.

(G) Durability, Wear Resistance, etc.

It is frequently claimed that a coupling has no wearing parts and that it will last as long as the prime mover to which it is coupled. From the viewpoint of the user, it is desirable to look into these statements more closely. In the case of all-metal couplings, the load-carrying components will show little or no wear if properly lubricated ; this calls for periodic inspection and checking. All-metal disc couplings are probably unique in not requiring any lubrication, a great advantage in cases where they can be suitably employed. Fatigue without apparent signs of wear can, of course, occur in all-metal couplings, since these, like other couplings, are subjected to bending stresses. Rubber couplings may wear, either owing to gradual fatigue of the rubber under shear, tension, or compression, or owing to friction against metal surfaces. Wear of mating parts also occurs, to some extent, in metal couplings after a long period of operation.

With regard to wear, two aspects to be borne in mind are :—(1) If the parts (e.g., rubber bushes) are easily replaceable during normal shut-down, it is of little importance whether they wear or not, provided that they

List of Particulars to be Considered in the Selection of Flexible Couplings

(1) Minimum and maximum operating speeds.

(2) Normal and maximum power to be transmitted.

(3) Maximum torque to be transmitted.

(4) Actual diameter, or nominal diameter with limits, of coupling shafts, and lengths of shaft extensions.

(5) Particulars of keyways or other arrangements for assembly.

(6) Type of prime mover.

(7) Type of driven machine.

(8) Maximum expected amounts of misalignment (parallel, angular, and axial).

(9) Position of shafts to be connected (horizontal, vertical, or inclined), and position of bearings in relation to the coupling halves.

(10) Space available for dismantling.

(11) Operating conditions (temperature, dust, oil, corrosive atmospheres, etc.).

(12) Driven machine direct-coupled or gear-driven ; reversible or non-reversible drive.

(13) Moment of inertia and position of any flywheel in the drive.

(14) Type of load (uniform, pulsating, intermittent, impact, etc.).

(15) Estimated cyclic irregularity of the drive.

(16) Special requirements for torsional-vibration limitation :—

- (a) Torsional stiffness of coupling.
- (b) Moment of inertia of the two main coupling members.
- (c) Desirability of vibration reduction due to damping or detuning action (variable stiffness) of the coupling.
- (d) Estimated maximum vibratory inertia torque to which the coupling will be subjected.

(17) Possibility of continuous axial movement of shaft ends, i.e., axial vibration.

(18) Special requirements for severe operating conditions :—

- (a) Maximum torques during starting and stopping.
- (b) Maximum pulsating torque or peak torque occurring during load changes.
- (c) Maximum impact torque, if impacts are expected to occur under normal operating conditions.
- (d) Provision, if necessary, of overload-protection or torque-limiting devices.

have a reasonably long service life ; and (2) replacement of individual components is acceptable if they impart special properties to the coupling, e.g., particularly high flexibility, small size, or low cost.

It may also be noted that wear resistance and the requirement for little or no maintenance and supervision are considerations arising primarily in the case of couplings which are designed to take appreciable torques or to operate at fairly high speeds, or which are otherwise subjected to severe running conditions. The amount of wear becomes practically immaterial in fractional-horsepower drives at moderate speeds. Nevertheless, even in such applications it is advisable to verify that the torque is uniform or varying only slowly ; otherwise, wear again becomes an important factor.

The question of wear is associated with the load-carrying capacity of the coupling. If the coupling is of ample size, and if it is operating within its permissible limits for misalignment, impact, etc., it can be expected to show least wear, even after a long period in service. In this connection, it cannot be overstressed that it is not necessarily the cheapest coupling, but the most suitable as regards properties and size, which will give the most economic service in the long run.

In difficult cases, satisfactory operation, durability, and minimum wear of a coupling can be ensured only if its selection is based on thorough consideration, by both user and manufacturer, of all operating conditions likely to be encountered. The list given on the previous page should serve as a general guide regarding the type of information which may have to be supplied by the user to the coupling manufacturer. In simple and straightforward applications, however, it will usually be sufficient to confine attention to the first seven or eight points in the list.

(H) Comparative Ease of Assembly and Disassembly

For the testing of mass-produced vehicle engines on dynamometer test beds, it is desirable to be able to connect and disconnect engine units fairly rapidly. Similar requirements exist for the routine production testing of small motor- or engine-driven generators, pumps, and other auxiliaries. For such applications, as well as in cases where periodic replacement or maintenance is required, it is an advantage to provide couplings which can easily be assembled and disconnected. Many couplings are designed with this requirement in mind ; others are available in special versions for ease of assembly. In fact, most couplings can be modified to fulfil this purpose to some extent.

When dismantling a coupling, keyed hubs should preferably remain undisturbed, since properly fitted keys can be damaged by frequent removal and refitting. Similar considerations apply, of course, to press and shrink fits.

(I) Operation in Either Direction of Rotation

Practically all permanent-engagement couplings, as distinct from many disengaging couplings or clutches, can be used in either direction of rotation. In vehicle and marine installations, as well as in machine tools, etc., a reversing drive is employed. In such cases, it is recommended practice to use a coupling of somewhat greater torque-carrying capacity than for a unidirectional drive, the rating being based on an increase of, say, 10 to 20% in the safety factor *S*.

For rapid reversal, the coupling should preferably have minimum wind-up (rubber couplings) or minimum backlash (gear and bush couplings). Neither of these requirements is of importance, however, for most spring couplings, metal disc couplings, rubber block couplings, and rubber disc couplings. Nevertheless, it is always advisable to specify rapid-reversal requirements to manufacturers, indicating, if possible, the approximate duration of the reversal and the inertias of the masses connected to the coupling during reversal.

Classification and Descriptions of Couplings

THE first feature to be considered in the classification of flexible couplings is size. For any prespecified type of coupling, size usually increases with torque-carrying capacity ; however, a comparison of various couplings evolved from similar basic considerations shows that some are more compactly designed than others, though this may restrict their range of operating conditions. On the other hand, kinematically speaking, many couplings may be regarded as equivalent, but, in practice, a classification based on kinematic principles alone would tend to be too theoretical for reference in dealing with practical requirements.

The simplest features which are both typical and suitable for classification purposes and ease of reference are the materials used. It is a fact that soft, flexible materials are usually associated with low torque-carrying capacities, and stiffer or harder materials with high torque-carrying capacities, so that it is convenient to classify flexible couplings into two basic groups, i.e. (1) couplings incorporating non-metallic components, e.g., rubber, fabric, leather, cork, etc., and (2) couplings consisting solely of metal components.

It is, of course, necessary to consider detail features by means of a suitable sub-classification, with indications of particular properties, advantages, and limitations. The classification adopted for this purpose is as follows :—

(1) Couplings Incorporating Non-Metallic Components

(1a) Sleeve or hose couplings.
(1b) Disc or sandwich couplings, unbonded and bonded.
(1c) Bush or pin couplings, with and without linkage systems.
(1d) Rubber block, spider, radial rubber-spring, and modified Oldham couplings.
(1e) Flexible ring or link couplings.
(1f) Belt and rope couplings.

(2) All-Metal Couplings

(2a) Metal disc couplings.
(2b) Metal spring couplings.
(2c) Metal bush or pin couplings.
(2d) Gear couplings.
(2e) Chain couplings.

Generalizations are naturally very often pointless, and no attempt can therefore be made in this survey to indicate any preference for either class of flexible coupling. Rather than make comparisons, it may be pointed out that both types of couplings have in some respects complementary properties, and, in the variety of practical requirements, selection should be based on the extent to which different couplings meet individual cases.

It should be noted that, in the following descriptions of couplings, the power ratings given are in all cases the theoretical maximum, i.e., for use under ideal conditions, so that they must be modified accordingly by the safety factors recommended by the manufacturers for the particular application concerned.

COUPLINGS INCORPORATING NON-METALLIC COMPONENTS

(a) Sleeve or Hose Couplings

The simplest couplings of this type consist of a length of rubber or rubberized-fabric tube, into which the chamfered ends of the driving and driven shafts are fitted, as shown in Fig. 2, the sleeve having an inner diameter somewhat smaller than that of the two shafts. This type of coupling is suitable for light torques and can take up a certain amount of angular, radial, and axial misalignment. The damping introduced is very small, but it will slip or shear under excessive shock loads, and it therefore affords some degree of protection against overloads.

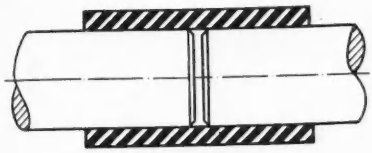


Fig. 2. Simple type of hose coupling.

The rubber sleeve may also be fixed to the shafts by means of a suitable rubber-base adhesive, giving the coupling a somewhat increased torque-carrying capacity. In this case, the coupling will not slip, but will fail in shear or torsion under excessive torques. It will still accommodate considerable degrees of misalignment, particularly if there is an appreciable free length of rubber hose between the ends of the shafts.

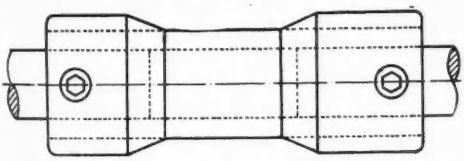


Fig. 3. Hose coupling with rubber held by collars and grub-screws.

Other simple designs in this category consist of a rubber hose held on the shaft portions by means of collars and grub-screws (Fig. 3) or by means of hose-clamps. In some cases, i.e., for shafts of comparatively large diameter, it is convenient to use a split sleeve, made of rubber, rubberized fabric, or leather, which is secured to the shafts by hose-clamps, as shown in Fig. 4, this type of coupling being suitable for low torques. The greater the gap between the ends of the shafts, the greater will be the amounts of misalignment which can be accommodated by such couplings. These sleeve arrangements have a low torsional stiffness and are mainly suitable for low or moderate operating speeds.

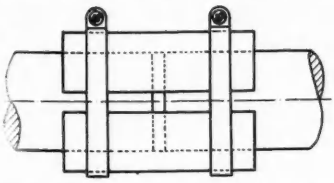


Fig. 4. Coupling with split sleeve secured to shaft by hose-clamps.

Commercially designed sleeve couplings are more compact or have a higher torque-carrying capacity than those mentioned above. A simple design of sleeve coupling, produced in the U.K. by Metalastik Ltd., in which the rubber sleeve is bonded to a flanged end, is shown in Fig. 5. This design, especially suitable for high-speed operation, provides high torsional flexibility and accommodates shock loads, vibration, and angular misalignment. Maximum power ratings are about 18 hp per 100 rpm. For greater flexibility, two of these couplings, separated by a light shaft, can be used in series.

Fig. 5. Coupling with sleeve bonded to a flanged end.
(Courtesy of Metalastik Ltd.)

The " Harrisflex " range of couplings basically consists of self-tightening male and female caps, threaded right- or left-hand, to suit the direction of rotation, and a rubber sleeve fitted into each cap. The coupling is completed by the addition of the necessary spacers. This design (Fig. 6), which is produced in the U.K. by Howard Clayton-Wright Ltd., is available in four types, covering maximum torques ranging from 4 lb-in. for a $\frac{3}{4}$-in. coupling up to 825 lb-in. for a 3-in. coupling.

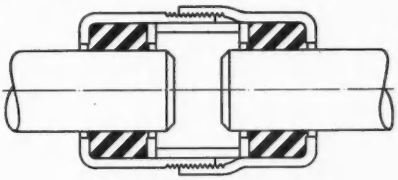


Fig. 6. " Harrisflex " sleeve coupling.
(Courtesy of Howard Clayton-Wright Ltd.)

The " Clayflex " split-sleeve coupling (Fig. 7), developed by Howard Clayton-Wright, consists of two inner members, suitably bored and keywayed, and an outer shell in two halves. The rubber is chemically bonded to the inner members and to the split outer

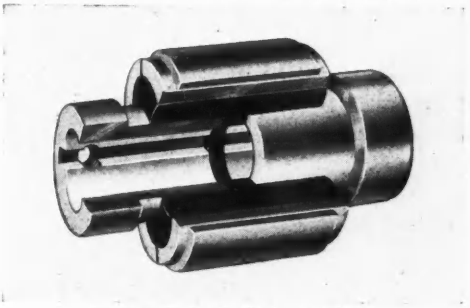


Fig. 7. " Clayflex " split-sleeve coupling.
(Courtesy of Howard Clayton-Wright Ltd.)

member in such a way as to leave axially disposed gaps between the halves of the outer shell. Finally, the outer clamping ring is forced over the outer shell, thus imposing a pre-stress on the rubber, which also separates the two inner members by its resulting axial deformation. In this coupling, which is intended to cater for a maximum torque of 630 lb-in., i.e., 10 hp at 1000 rpm, increased capacity for conditions of misalignment at reduced torques can be obtained by variations in the thickness of the rubber wall, while the amount of torsional deflection can similarly be varied by wall thickness and by alterations in the rubber compound used.

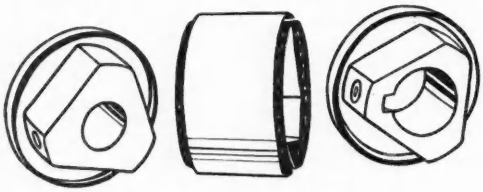


Fig. 8. " Flex-Ring " sleeve coupling.
(Courtesy of Smith & Serrell Inc.)

To permit some axial movement, the rubber sleeve can be arranged as a bush inside a metal sleeve. If the bush is moulded with a triangular-shaped hollow centre, as in the " Flex-Ring " coupling (Fig. 8), it can be fitted over correspondingly shaped spigots on end-covers keyed to the shafts. This design, made in the U.S.A. by Smith & Serrell Inc., is particularly suitable for fractional-horsepower applications and permits appreciable axial movement and angular misalignment, but reduces the amount of permissible parallel misalignment. Another advantage is that it is easily dismantled.

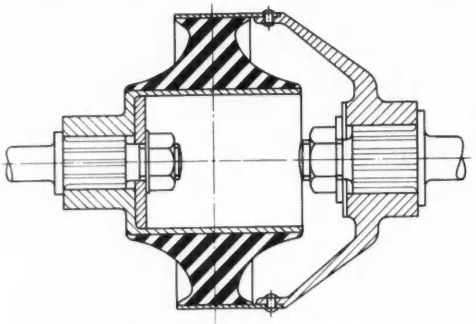


Fig. 9. " Flexilant K " sleeve coupling.
(Courtesy of Rubber Bonders Ltd.)

The " Flexilant K " sleeve coupling (Fig. 9), produced in the U.K. by Rubber Bonders Ltd., is, in effect, a disc of hyperbolic contour, bonded to two tubular metal parts. Small sizes have been developed for aero-engine auxiliaries and larger sizes for blowers, pumps, etc. Owing to its variable stiffness, this coupling has some detuning action. Maximum torsional deflection is 35 deg. in the smaller sizes, while the largest sizes can transmit torques up to 50,000 lb-in. These couplings permit angular misalignment up to 30 deg. (small sizes) and parallel and axial misalignments of $\frac{1}{8}$ and 1 in. respectively.

In another design, the moulded-rubber sleeve is provided with internal splines (Fig. 10), and splined end-castings fit into the splines of the rubber sleeve. This arrangement is suitable for overload protection since, under excessive torques, only the rubber splines will be destroyed. This coupling accommodates some misalignment and axial movement, and can transmit moderate amounts of power (up to about 25 hp). The particular design illustrated in Fig. 10 is made in the U.S.A. by the Guardian Products Corporation.

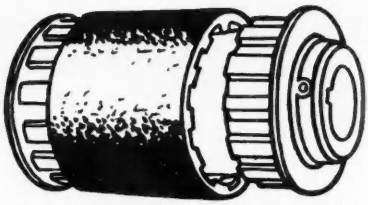


Fig. 10. Moulded-rubber sleeve coupling with internal splines.
(Courtesy of Guardian Products Corporation)

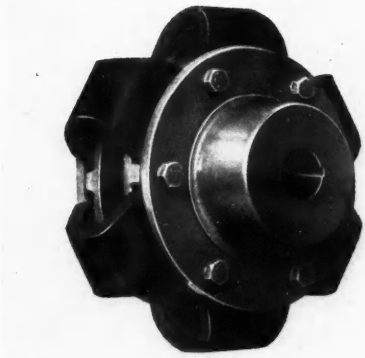


Fig. 11. Peripheral-strip " Multicross " coupling.
(Courtesy of Dipl.-Ing. Herwarth Reich)

Further developments of the rubber-sleeve principle have led to rubber-strip and rubber-tyre types of couplings. The peripheral-strip " Multicross " coupling (Fig. 11), made in Germany by Reich, is highly flexible, and accommodates very considerable parallel, axial, and angular misalignments. In this design, torque is trans-

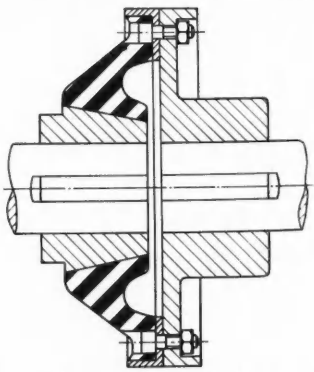


Fig. 12. " Kegelflex " bonded rubber flange coupling.
(Courtesy of Lohmann & Stolterfoht)

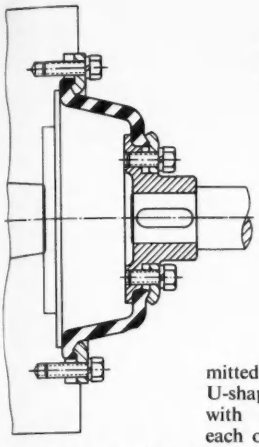
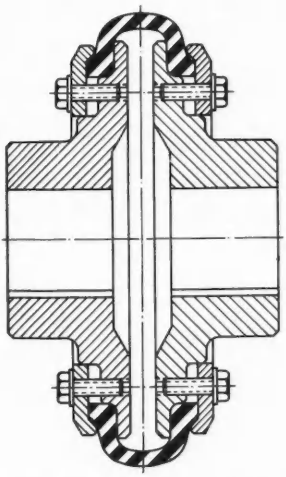


Fig. 13. " Periflex " clamped rubber flange coupling.
(*Courtesy of Maschinenfabrik Stromag*)

mitted by means of special U-shaped rubber cushions with cord slings crossing each other in the shape of a figure-8, cams on the hub flange gripping in recesses on the inner sides of the legs of the rubber cushions. Power ratings of these couplings, which are easily dismantled and which, in addition, exercise a certain damping effect on vibration and absorption of shock loads, range from about 15 hp at 5000 rpm for a $3\frac{3}{4}$-in. coupling to 1300 hp at 650 rpm for a 35-in. coupling.

Higher torque-carrying capacity is obtained with the " Kegelflex " bonded rubber flange type of coupling shown in Fig. 12. In this design, the rubber sleeve or flange is bonded to a metal ring connected by means of bolts to a flange and hub on the driven shaft. These couplings accommodate parallel, axial, and angular misalignments (up to 4 deg.). Where even greater flexibility is required, two hub-and-flange sections can be bolted together. The maximum torsional deflection of this coupling, which is marketed in Germany by Lohmann & Stolterfoht, is up to 20 deg. Instead of being bonded. the rubber flange can, of course, also be secured by means of clamping arrangements, as in the "Periflex" design illustrated in Fig. 13.

The " Periflex " rubber-tyre type of coupling (Fig. 14), which is made in Germany by Stromag and which is at present available in England from Stone-Wallwork Ltd., basically consists of a peripheral sleeve or tyre which is transversely slit, slipped over its supporting flanges, and held in position by means of outer ring sections



ig. 14. " Periflex " rubber-tyre shaft coupling.
(*Courtesy of Maschinenfabrik Stromag*)

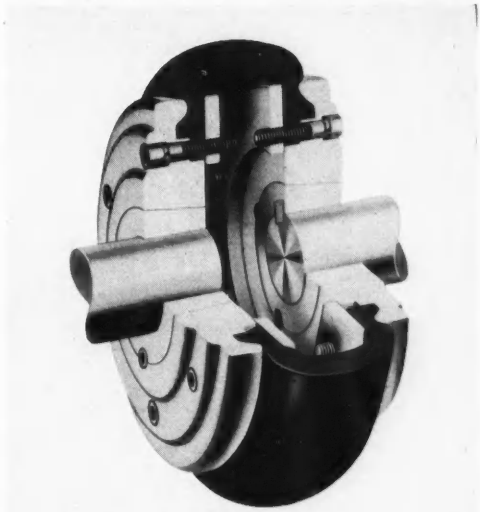


Fig. 15. Cut-away view of the " Para-flex " rubber-tyre type of flexible coupling.
(*Courtesy of Dodge Manufacturing Corp.*)

secured to the flanges by bolts. The sleeve is easily replaceable, and standard couplings of this type have comparatively high torsional flexibility, with rated powers from about 3 to 1100 hp and maximum speeds from 800 to 4000 rpm. In addition, " Periflex " couplings are available for special low- and high-speed applications, and with a specially shaped tyre for particularly large angular and axial misalignments. A similar coupling, known as the " Para-flex ", and made in the U.S.A. by the Dodge Manufacturing Corporation, is shown in Fig. 15. A cut-away section of the " Para-flex " tyre, showing the special tension members incorporated, is illustrated in Fig. 16.

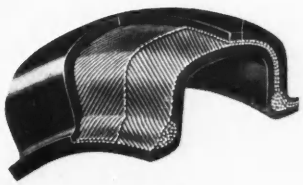


Fig. 16. Cut-away section of " Para-flex " coupling tyre, showing special tension members.
(*Courtesy of Dodge Manufacturing Corp.*)

Relatively high torque-carrying capacity is also provided by the rubber-sleeve type of couplings shown in Figs. 17 and 18. The " Double-Cone " coupling (Fig. 17), made in Great Britain by Metalastik Ltd., consists of two rubber annuli or sleeves which are bonded on both sides to metal sleeves. The inner and outer metal sleeves are

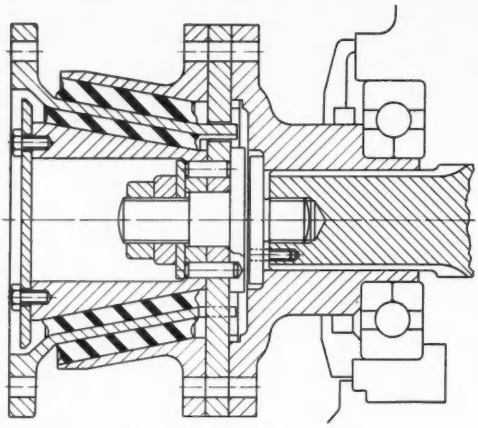


Fig. 17. " **Double-Cone** " **coupling.**
(*Courtesy of Metalastik Ltd.*)

bolted to the driven shaft, whereas the intermediate metal sleeve is coupled to the driving shaft. This " parallel-spring " arrangement gives increased torque-carrying capacity. During assembly, the inner sleeve is drawn axially into the cone, thus precompressing the rubber and giving the coupling sufficient radial stiffness to make it self-centering, so that one of the shafts does not need a bearing close to the joint. Radial stops are provided and these serve to prevent overloading of the rubber under excessive torques, the maximum permissible torque being 100,000 lb-in. on the largest model, corresponding to an angular twist of 10 deg. This coupling accommodates some angular and parallel misalignment. Speeds in excess of 8000 rpm are permissible, depending on size and application. The " Double-Cone " coupling is mainly used in diesel locomotives, lorries, and motor launches, between the gearbox and the cardan- or propeller-shaft, as well as for the gearbox output shafts of high-speed turbines ; it is also used in pressure-charger and blower drives.

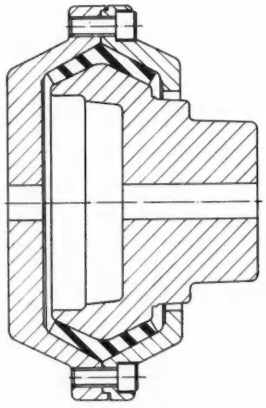


Fig. 18. " **Bicone** " **coupling.**
(*Courtesy of Metalastik Ltd.*)

In the " Bicone " coupling, also made by Metalastik (Fig. 18), the rubber is bonded on its inner surface to the driven member, while its outer surface, by suitable design, is pressed against the concave surface of the driving member. This coupling incorporates safety stops, to maintain a form of drive in case of damage to the coupling under excessive loads. It is comparatively flexible (torsional stiffness is of the order of 5700 lb-ft/rad.) and can take an axial end-load of 3000 lb, with an axial movement of 0·08 in.

Although it is in effect a clutch, the Fawick " Airflex " is also a flexible coupling. This design, which is marketed in England by Crofts (Engineers) Ltd. and in Germany by Lohmann & Stolterfoht, basically comprises two concentric members, one connected to the driving shaft and the other to the driven shaft, contact between these members being established by a rubber tyre inflated with compressed air to a pressure of 70 to 100 psi. The high radial flexibility of this unit, which is particularly suitable for rough and heavy drives, enables it to accommodate a substantial degree of parallel misalignment. Rated torques are from about 1800 to 240,000 lb-in., with maximum powers from 84 to 2000 hp and speeds from 500 to 2800 rpm.

(b) Disc or Sandwich Couplings, Unbonded and Bonded

In its simplest form, the unbonded disc coupling consists of a rubber, fibre-and-rubber, fabric-and-rubber, or leather disc connected by means of alternating sets of pins, studs, or bolts to two carrier members fixed on the driving and driven shafts. An example of such a disc is the " Hardy " disc (Fig. 19), made in the U.K. by Hardy, Spicer & Co. Ltd. This disc is constructed of laminated fabric, using a staggered principle, so that the threads of the various layers run in all directions and ensure that the tensile strength of the fabric between each pair of bolt holes is identical. For greater strength, steel plates are riveted on either side of the bolt holes.

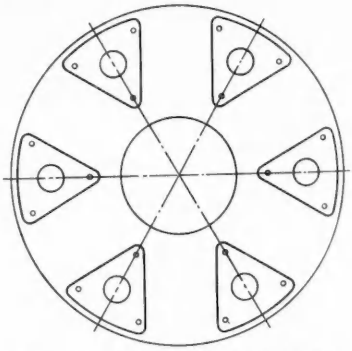


Fig. 19. " **Hardy** " **laminated-fabric flexible disc.**
(*Courtesy of Hardy, Spicer & Co. Ltd.*)

Another disc in this category is the " Hermetic " flexible fabric disc, made in Great Britain by the Hermetic Rubber Co. Ltd. Constructed of rubber-impregnated cotton fabric, these discs are jig-moulded to ensure accuracy, and vulcanized by a special method intended to confer maximum tensile strength and flexibility. These discs may be fitted with divided ferruled plates (Fig. 20), bolting pressure being automatically regulated by the engagement of the two opposing ends of the divided ferrules in the centre of the disc.

A flexible disc coupling, in which the disc is carried on steel pins mounted on flanged hubs and inserted into

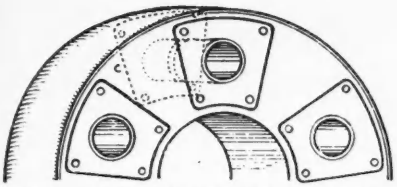


Fig. 20. " Hermetic " flexible fabric disc, with divided ferruled plates. (*Courtesy of Hermetic Rubber Co. Ltd.*)

slotted recesses, is produced in Great Britain by Renold Chains Ltd. In this design (Fig. 21), the discs are built up of staggered layers of rubber-impregnated fabric, and the pins, secured by lock-nuts, are a taper fit in the half-bodies, which are arranged for keying to the shafts. The maximum power ratings of these couplings, which can accommodate an angular misalignment of 2 deg. between shafts, range from 36 hp at 3600 rpm for a 4-in. coupling up to 600 hp at 1000 rpm for a 12½-in. coupling.

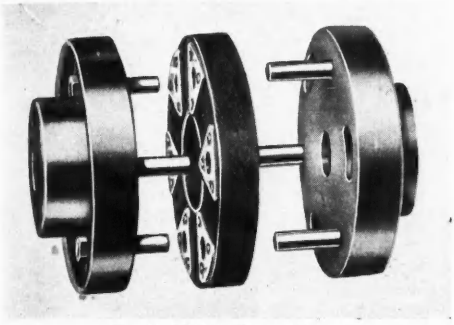


Fig. 21. Flexible disc coupling. (*Courtesy of Renold Chains Ltd.*)

Another example of an unbonded rubber disc coupling is the " Taper-Lock " coupling produced in Great Britain by J. H. Fenner & Co. Ltd., (Fig. 22). This design has a central disc of rubber-impregnated fabric mounted on steel pins, the flanges being fitted with high-grade cast-iron bushes having a saw-cut through one side only. The tapered bushes are drawn into the tapered bore of the flanged members by means of grub-screws, thus nipping the shaft. In addition, the bushes are keyed to the shafts by standard parallel keys. The power ratings of these couplings, which accommodate axial, angular, and some radial misalignment, are up to 150 hp at 1800 rpm, maximum speeds being up to 3600 rpm.

In addition to a range of flexible disc couplings of the type already described, Crofts (Engineers) Ltd. manufacture their " Perfect " coupling (Fig. 23). Essentially, this design consists of two specially formed aluminium-alloy die castings with projecting lugs, which engage in slots provided in a centre disc of vulcanized-rubber or chrome-leather construction. These couplings are also reversible and intended for applications involving low powers, i.e., up to 5 hp at 1000 rpm.

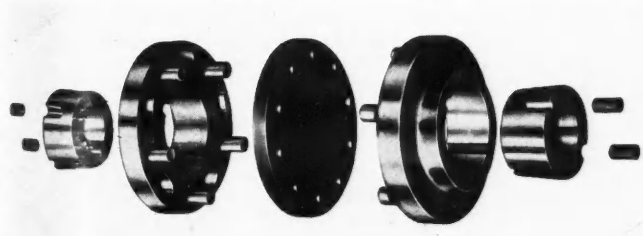


Fig. 22. " Taper-Lock " flexible disc coupling. (*Courtesy of J. H. Fenner & Co. Ltd.*)

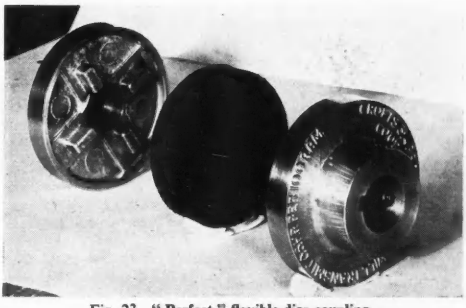


Fig. 23. " Perfect " flexible disc coupling. (*Courtesy of Crofts (Engineers) Ltd.*)

Another unbonded rubber disc coupling is shown in Fig. 24. This coupling, marketed in Germany by Lohmann & Stolterfoht, employs pins which are provided with shoulders to give end location, and has a comparatively high torsional stiffness, like other couplings of this type.

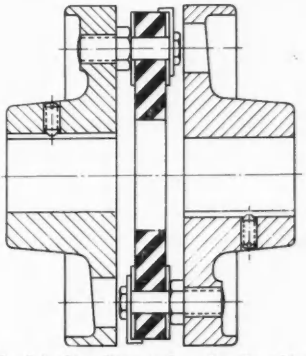


Fig. 24. Unbonded rubber disc coupling using pins with shoulders to give end location. (*Courtesy of Lohmann & Stolterfoht*)

A rubber disc coupling of specially moulded shape is illustrated in Fig. 25. This coupling, made in Germany by Heinrich Desch, consists of a conical solid-rubber moulding with peripheral notches for the steel pins. The hollow centre of the cone is splined and fits over a splined carrier which is keyed to the driven shaft, the pins being shrunk into the flange of the driving member. Power ratings range from about 25 hp at 7400 rpm for a 3½-in. coupling to 675 hp at 1500 rpm for a 17-in. coupling.

A simple type of bonded-rubber disc coupling is illustrated in Fig. 26. In this " Flexilant " design, made in the U.K. by

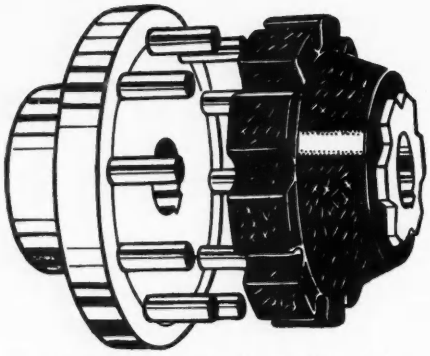


Fig. 25. **Rubber disc coupling with solid-rubber moulding.** (*Courtesy of Heinrich Desch G.m.b.H.*)

Rubber Bonders Ltd., the rubber disc or sandwich is bonded to flanged hubs. This coupling accommodates angular, parallel, and axial misalignment, has a maximum torsional deflection of 20 deg., and is suitable for light drives. Another simple type is the

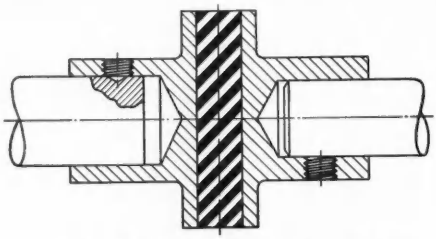


Fig. 26. **" Flexilant " bonded-rubber disc coupling.** (*Courtesy of Rubber Bonders Ltd.*)

"Silentbloc" bonded-type coupling (Fig. 27a), in which the two hubs are connected by a rubber element chemically bonded to them in the moulding process. This simple design also provides comparatively great flexibility and is also suitable for light drives, maximum power ratings at 1440 rpm being about $\frac{1}{8}$ hp for a $\frac{3}{4}$-in. coupling to 1 hp for a $1\frac{1}{2}$-in. coupling. A bonded disc coupling in which the

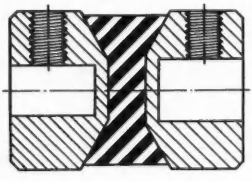


Fig. 27a. **"Silentbloc" bonded-rubber disc coupling.** (*Courtesy of Silentbloc Limited*)

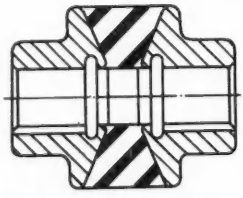


Fig. 27b. **Coupling with tapered sleeve bonded to metal hubs.** (*Courtesy of Metalastik Ltd.*)

disc has a taper and is bonded to suitably shaped metal hubs is shown in Fig. 27b. This coupling, which is made in the U.K. by Metalastik Ltd., absorbs shocks and damps out vibrations, possesses high torsional flexibility, and accommodates shaft misalignment.

This group may be considered to include "barrel-type" couplings, such as the design (Fig. 28) produced in Great Britain by Dunlop. This coupling consists of a moulded-rubber centre-section which has an external taper and which is bonded to its hub portions; it can accommodate parallel misalignment up to $\frac{1}{8}$ in., angular misalignment up to 5 deg., and axial strains up to 20% in compression and 5% in tension, and torsional deflections from 18 to 60 deg., depending on size. By using a number of rubber compounds of different stiffnesses, it is possible to extend the torsional-deflection values to cover a range of about 10:1 for each size of coupling. These couplings are designed for fractional- and low-horsepower applications, torque ratings being from 3·2 to 340 lb-in., with maximum speeds up to 5000 rpm. Above this speed, torque-carrying capacity may be reduced, to allow for the stresses in the rubber caused by centrifugal force.

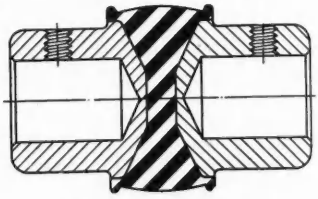


Fig. 28. **Barrel-type coupling.** (*Courtesy of Dunlop Rubber Co. Ltd.*)

Of other bonded-type " Flexilant " couplings made by Rubber Bonders Ltd., the design illustrated in Fig. 29 incorporates a rubber disc in a flywheel. With this design, a wide range of grades of rubber permits selection of the required stiffness for each particular application, rated powers being up to 60 hp and speeds up to 2000 rpm, with maximum transmitted torques of 6000 lb-in.

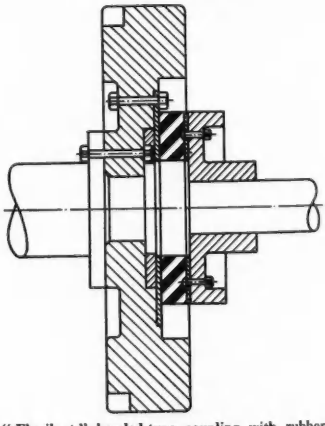


Fig. 29. **" Flexilant " bonded-type coupling with rubber disc in a flywheel.** (*Courtesy of Rubber Bonders Ltd.*)

A bonded-rubber disc coupling is shown in Fig. 30, in which the rubber disc is bonded to steel plates connected to hub members by bolts, the hubs being keyed to the shafts. In this design, which is made in England by the Dunlop Rubber Co. Ltd., and by R. Hunt & Co. Ltd., the moulded-rubber disc incorporates holes through

which the bolts are fitted to the flanged hubs. This coupling accommodates parallel, angular, and axial misalignments, and torsional stiffness is guaranteed to within fine limits. Maximum end-float is normally restricted to a 20% compressive or 5% tensile strain on the thickness of the rubber disc (e.g., a coupling of 0·875 in. thickness can accommodate 0·175 in. end-float in compression and 0·044 in. in tension). The power ratings of these couplings are from about 50 hp at 6000 rpm for a 4-in. coupling up to 1450 hp at 1500 rpm for a 19½-in. coupling.

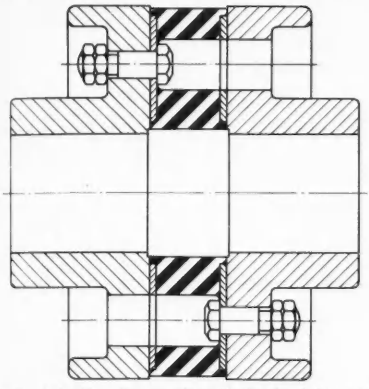


Fig. 30. **Bonded-rubber disc coupling with disc bonded to steel plates connected to hub members by bolts.**
(*Courtesy of Dunlop Rubber Co. Ltd.*)

A V-section rubber disc coupling is shown in Fig. 31. By virtue of this V-section, this design, which is also produced by Rubber Bonders Ltd., permits a more uniform distribution of torsional stress throughout the rubber than is the case with parallel disc designs. This type of " Flexilant " coupling, which is suitable for high-speed marine drives, accommodates angular, parallel, and axial misalignments, and has power ratings up to about 580 hp and maximum speeds of 2800 rpm. Maximum torsional deflection is 25 deg.

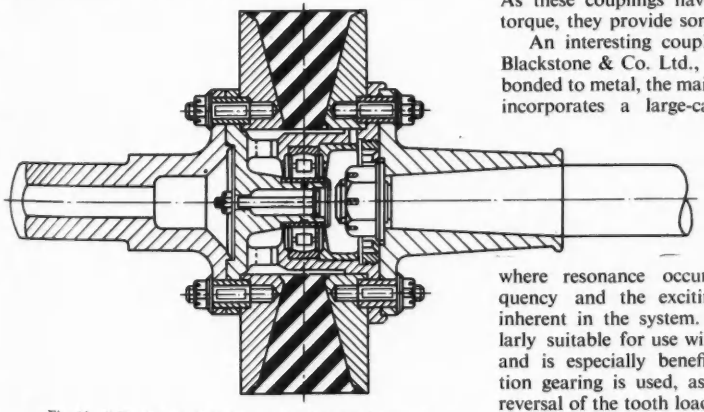


Fig. 31. **" Flexilant " V-section rubber disc coupling.**
(*Courtesy of Rubber Bonders Ltd.*)

In addition, Rubber Bonders Ltd. have produced a series of multiple-disc couplings, one of which is shown in Fig. 32. This particular unit, the " Flexilant C-300 " (heavy-duty type), is designed to transmit considerable torque, while still retaining a high degree of flexibility. For this purpose, a number of flat (parallel) or V-shaped rubber discs is assembled into a single coupling, the discs being bonded on both sides to intermediate or end-plates connected alternately to the driving and driven hub portions of the coupling. The distance-pieces are waisted at their centres, to allow angular movement of the centre-plate without metal-to-metal contact, the metal projections of this plate (which has a " scalloped " contour) making contact with the distance-pieces when the coupling is grossly

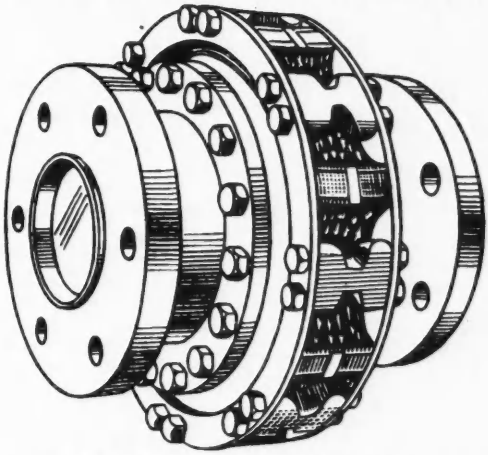


Fig. 32. **" Flexilant " heavy-duty multiple-disc coupling.**
(*Courtesy of Rubber Bonders Ltd.*)

overloaded. This is desirable, for instance, in marine installations, where the coupling must at all costs continue to function. Maximum torques up to 250,000 lb-in., with torsional stiffnesses between $0{\cdot}12 \times 10^6$ and $4{\cdot}0 \times 10^6$ lb-in./rad., are quoted for couplings of this type. Maximum torsional deflection is 15 deg., and some angular, parallel, and axial misalignment can be accommodated. As these couplings have a stiffness which varies with torque, they provide some detuning action.

An interesting coupling, produced in the U.K. by Blackstone & Co. Ltd., consists of two units of rubber bonded to metal, the main feature being that the coupling incorporates a large-capacity viscous-fluid torsional-oscillation damper, the object being to smooth out cyclic variations in torque between the driving and driven units and to eliminate any dangerous torsional oscillations which may develop at speeds where resonance occurs between the natural frequency and the exciting impulses which may be inherent in the system. This arrangement is particularly suitable for use with internal combustion engines and is especially beneficial when increasing or reduction gearing is used, as it smooths out and prevents reversal of the tooth loading.

(c) Bush or Pin Couplings, with and without Linkage Systems

In order to provide some flexibility, it is evident that rigid couplings can be modified by replacing the coupling bolts with rubber or leather pads, which must, of course,

be of greater cross-section than the coupling bolts, to sustain the same shear forces. In a typical example of a coupling developed in accordance with this principle, leather pads, of considerable peripheral width, are inserted into a central metal disc and fit in recesses provided in flanged hub members. In other designs, the pads are of circular cross-section and are fitted in the form of bushes on bolts or rivets.

Nowadays, most bush or pin couplings incorporate rubber or leather bushes specially designed and manufactured for the purpose. They may take the form of plain rubber bushes, or the rubber may be bonded to a metal sleeve or may have a fabric lining. Some rubber bushes are also provided with an external metal sleeve, so as to resist abrasion caused by axial movement in the holes of the flanged coupling members. Still other bushes are made up of a number of rubber rings of conical section, providing greater space for deformation of the rubber than plain cylindrical bushes. In addition, some bushes have concave end-sections, permitting the rubber to flow in the direction of the bush axis under load, while certain types are bonded on their inner and outer surfaces to metal sleeves and are used not only to sustain compressive loads (with some shear flow) but also to serve as " torsion-bars ", i.e., they can be used in link-type or articulated couplings to obtain a certain amount of restoring force.

From three to twenty bushes or " buffers " are generally used, depending on the size of the coupling and the torque-carrying capacity required, the bushes being mounted on steel pins and inserted into holes in one of the two flanged members. In most cases the bushes are made of a rubber compound, though leather bushes are sometimes employed.

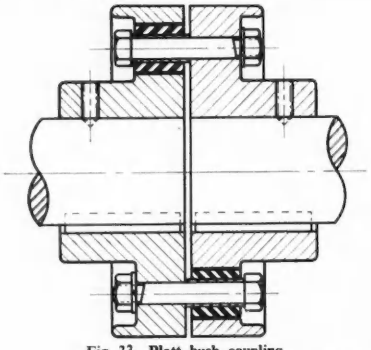


Fig. 33. Platt bush coupling.
(Courtesy of J. H. Fenner & Co. Ltd.)

Bush- or pin-type couplings have enjoyed great popularity for many years, and are produced by a very large number of manufacturers, e.g., Platt, Crofts, Silentbloc, David Brown, Hunt, Williams & Womersley, etc. in Great Britain.

In the Platt coupling (Fig. 33), both halves are identical, the metal-housed rubber bearings being driven into and equally disposed in each half, whilst the holes for the connecting pins or bolts are arranged alternately between them. If desired, one half only of the coupling may be used, connected to suitable holes in a flywheel, existing flange adaptor, etc. The power ratings of this coupling range from about 160 hp at 5100 rpm for a 4½-in. coupling up to 2650 hp at 1300 rpm for an 18-in. coupling.

An example of a bush-type coupling, manufactured in the U.K. by Crofts (Engineers) Ltd., and in this case

Fig. 34. Bush-type coupling fitted with "Taper-Flushbush" (shown in detail on left).
(Courtesy of Crofts (Engineers) Ltd.)

fitted with their " Taper-Flushbush ", is shown in Fig. 34. Here, the flanges of the coupling have a tapered bore to accommodate the Flushbush, and are drilled and tapped to receive jackscrews which, tightened alternately, drive the Flushbush into the flanges. As the Flushbush is split on one side throughout its entire length, it grips the shaft sufficiently tightly to deal with any load to be transmitted. Dismantling and re-assembly of the coupling is facilitated, as partial withdrawal of the jackscrews is sufficient to release the grip on the shaft.

Fig. 35. " Cone-Ring " bush coupling.
(Courtesy of The David Brown Corp. (Sales) Ltd.)

A heavy-duty " Cone-Ring " coupling with double-taper or conical rubber rings, manufactured in the U.K. by David Brown, is shown in Fig. 35. In this design, a series of rings of conical section is used, instead of parallel bushes. Fig. 36a shows (with some exaggeration) the change in shape of these cone-ring bushes under parallel misalignment, in which the material is free to

deform in all directions. An additional feature is the freedom of axial motion provided by these bushes. End-float can take place with very little internal distortion producing stress and therefore axial resistance, as shown in Fig. 36b. The power ratings of these couplings are from about 20 hp at 6550 rpm for a 3½-in. coupling up to 13,000 hp at 500 rpm for a 45-in. coupling.

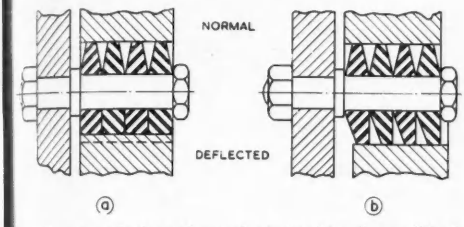


Fig. 36. **Change in shape of cone-ring bushes (a) under parallel misalignment, and (b) under axial motion.**

A rubber bush-type coupling, in this case with a floating disc, is shown in Fig. 37. This particular coupling, which is made in England by Silentbloc Ltd., comprises a centre member housing four equally spaced rubber blocks, alternately connected by bolts to one or other of a pair of flanged hub members, which are, in turn, fitted to the shafts to be coupled. The use of specially developed and shaped preloaded resilient rubber blocks provides great flexibility, and permits considerable deflection. A large amount of torsional vibration, as well as shocks and uneven impulses, can be absorbed. Furthermore, no lubrication for adjustment after fitting is required. Power ratings are from 0·15 hp per 100 rpm for a 2⅛-in. coupling to 20 hp per 100 rpm for a 12-in. coupling. Similar couplings are marketed in Germany by Boge & Sohn, in the U.S.A. by the Morse Chain Company (the "Morflex" coupling), and in the U.K. by The Laycock Engineering Co. Ltd. (the "Layrub" coupling).

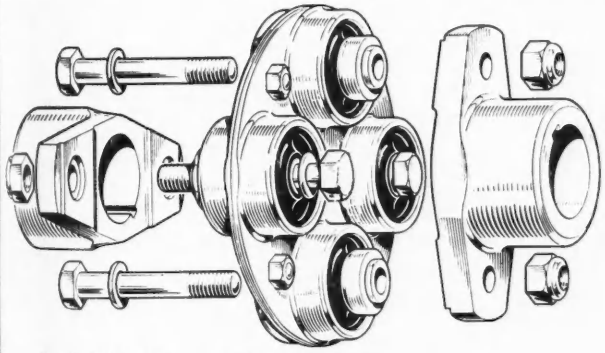


Fig. 37. **Rubber bush-type coupling with floating disc.** (*Courtesy of Silentbloc Limited*)

A coupling employing rubber "bobbins" is illustrated in Fig. 38. In this "Flexilant" design, made in the U.K. by Rubber Bonders Ltd., each rubber element or bobbin consists of a cylinder of rubber bonded to the enlarged heads of two metal studs, by means of which a number of bobbins can be bolted to the periphery of the flanged hubs of the coupling. For easy fitting and removal of the bobbins, the hub flanges are slotted radially. The bobbins are loaded in shear by the torque transmitted, but can also accept appreciable axial misalignment. The torsional stiffness of these couplings can be varied, as desired, by changing the number of bobbins or by using different grades of rubber compounds in their construction. Bobbin diameters range from ⅜ to 4 in. For couplings with 4-in. diameter bobbins, maximum torques of 72,000 lb-in. are possible. These couplings accommodate considerable angular, parallel, and axial misalignments, and permit maximum torsional deflections of 20 deg.

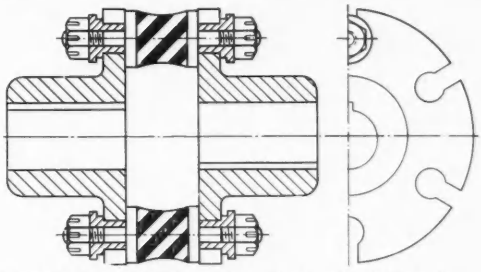


Fig. 38. **"Flexilant" bush coupling employing rubber "bobbins".** (*Courtesy of Rubber Bonders Ltd.*)

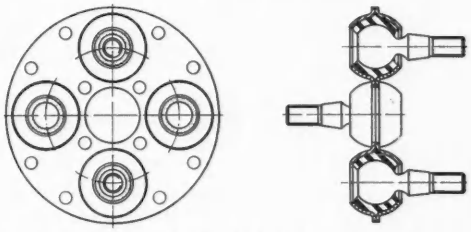


Fig. 39. **Bush coupling with rubber bushes bonded to the outer member.** (*Courtesy of Metalastik Ltd.*)

A rubber bush coupling which may be said to act as a universal joint is illustrated in Fig. 39. Produced in Great Britain by Metalastik Ltd., this coupling incorporates four ball-ended pins to which rubber is bonded, two pins being connected to the driving hub and two pins to the driven hub. The pivoting action of the pins permits this type of coupling to accommodate considerable angular misalignment, while maximum torques of 150 lb-ft and torsional stiffness of 5200 lb-ft/rad. can be obtained. Axial stiffness is of the order of 5000 lb/in. for the largest coupling of this type.

Couplings with radial bushes have also been developed. An example of a "Silentbloc" radial-type bush coupling is shown in Fig. 40. In this design, pins and rubber trunnion blocks are disposed radially, enabling loads to be taken in several directions. This feature is of particular importance in marine power transmission. These "Silentbloc" couplings, which are available in two models, for medium- and heavy-duty applications respectively, have power ratings from about 4½ hp per 100 rpm for a 6¾-in. coupling up to 500 hp per 100 rpm for a 30-in. coupling. Similar radial-type bush couplings are also

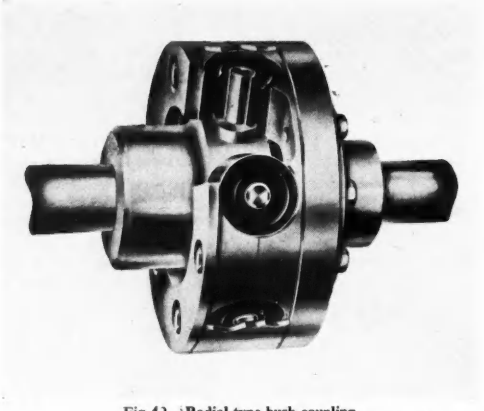


Fig. 40. Radial-type bush coupling.
(*Courtesy of Silentbloc Limited*)

produced by other manufacturers, e.g., by the Morse Chain Company in the U.S.A., whose " Morflex " radial couplings have power ratings from 3 hp per 100 rpm for a 4½-in. coupling to 262 hp per 100 rpm for an 18-in. coupling. In addition, the Morse Chain Company produce a range of radial couplings of similar type especially for marine purposes.

A bush-type coupling with linkage arrangements, also made by Silentbloc Ltd., is illustrated in Fig. 41. In this design, the bushes are arranged in pairs in separate link members, this type of construction facilitating connection and disconnection in confined spaces, in addition to accommodating shaft misalignment. The power ratings of this coupling are up to 26 hp per 100 rpm.

Another coupling with linkage arrangements is the " Trailing Link " coupling, made in England by Metalastik Ltd. (Fig. 42), which has a driving hub acting on one set of pins, while the driven hub acts on the other pins. This coupling, despite its comparatively short length, accommodates considerable misalignment ; thus, for example, a coupling, 5¾ in. in diameter and 2¼ in. between hubs, can accommodate 0·15 in. parallel misalignment and 3 deg. angular misalignment. Maximum power ratings of 20 hp per 100 rpm are possible.

The Twiflex articulated coupling is illustrated in Fig. 43. In this design, made in the U.K. by Twiflex Couplings Ltd., the driving and driven halves each have an equal number of arms, those of one half being coupled to those of the other half by articulated links. Each link system consists of a plain link with a renewable flexible bush pressed into each end and a forked link with a bush at the solid end. The bushes are made of special rubber and are bonded between inner and outer metal sleeves. The outer sleeves are pressed into the links, and the inner sleeves are gripped to the ends of the arms or between the limbs of the forks by through-bolts, so that all relative movement is confined to the rubber. The links are assembled " dog-legged ", so that, when torque is applied to the coupling, the linkages tend to straighten. As a result, the coupling has high torsional flexibility, with a stiffness that increases with increasing torque, thus altering the frequency of the system and detuning resonant torsional vibrations. These couplings, which are suitable for unidirectional drives only, can accommodate angular, parallel, and axial misalignments, and have maximum power ratings ranging from 2 hp per 100

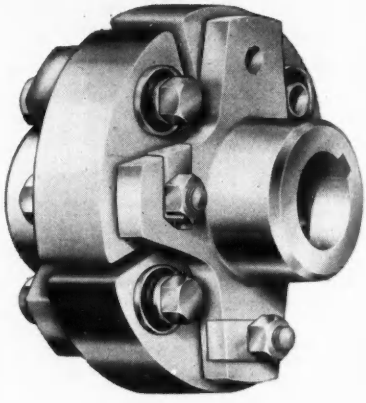


Fig. 41. Bush-type coupling with linkage arrangements.
(*Courtesy of Silentbloc Limited*)

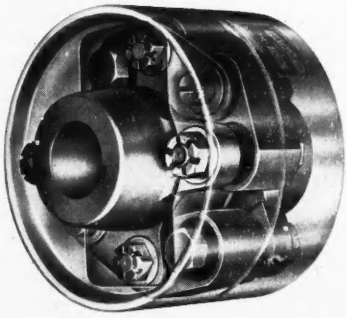


Fig. 42. " Trailing-Link " bush coupling.
(*Courtesy of Metalastik Ltd.*)

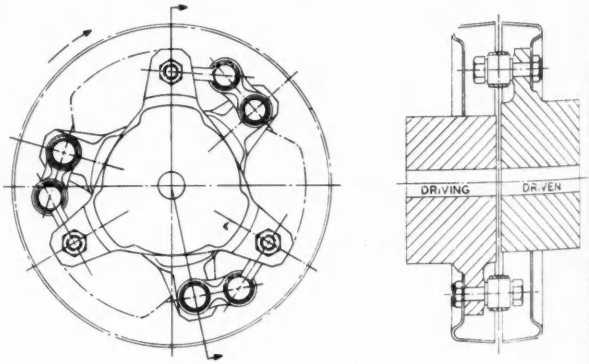


Fig. 43. Twiflex articulated coupling.
(*Courtesy of Twiflex Couplings Ltd.*)

rpm for a 6¼-in. coupling up to 50 hp per 100 rpm for a 20-in. coupling.

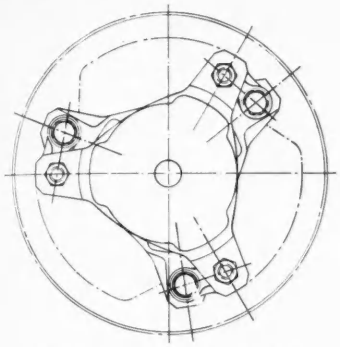


Fig. 44. **Twiflex " Flexilink " coupling.**
(*Courtesy of Twiflex Couplings Ltd.*)

The Twiflex " Flexilink " coupling (Fig. 44) is a simpler version of the Twiflex articulated coupling, using single plain links only, all components being identical with those used in the articulated coupling. It is much stiffer torsionally, but will accommodate appreciable angular misalignment. Owing to the link action with the two bushes in series, it is stated to be several times more flexible than the ordinary pin type of coupling, and allows considerable end-float. It can be used in both unidirectional and reversing drives, and is especially indicated where initial alignment is reasonably good, where little distortion is expected, and where full damping of torsional shocks and vibrations is not required. Maximum power ratings of the standard range are from 2 hp per 100 rpm for a 5¾-in. coupling up to 50 hp per 100 rpm for an 18½-in. coupling. In addition, Twiflex manufacture a range of automatic clutch couplings but these, although fully flexible, are basically intended as clutches and are therefore beyond the scope of this survey.

Rubber bushes are also employed in flexibly mounted gear-wheels. Units of this type are, for instance, made in the U.K. by the Metropolitan-Vickers Electrical Co. In such arrangements, the coupling becomes an integral part of the gear-wheel, resulting in a compact design.

(*d*) *Rubber-Block, Spider, Radial Rubber-Spring, and Modified Oldham Couplings*

Rubber-block couplings may be regarded as a development of dog-type couplings which have been given a certain amount of flexibility by fitting rubber blocks or pads between the dogs. In the simplest form of coupling of this type, the pads, of comparatively hard rubber, are bonded to one of the lateral faces of the dogs, the two coupling members being made of steel or of some strong synthetic material, such as Tufnol. Small couplings of this type are suitable for cam drives, etc. intended to sustain appreciable pulsating torques, but they are also made in considerably larger sizes for use as heavy-duty couplings. They eliminate the chatter caused by the normal metal contact of ordinary dog couplings and, besides providing some cushioning effect against shock loads, can accommodate axial, as well as parallel and angular, misalignment, if the dogs are not limited by contact with the end-faces of the flanged portions. A range of couplings of this type is produced by manufacturers all over the world, normally for low and moderate power ratings.

Double-row rubber-block couplings are also available. Couplings of this type consist of two rows or sets of rubber blocks with an intermediate metal member provided with dogs on both sides. The blocks have a rounded contour, permitting increased deformation under compressive loads. An example of this type of coupling is made in the U.S.A. by The Lovejoy Flexible Coupling Co.

A heavy-duty rubber-block coupling, made in the U.K. by The Holset Engineering Co. Ltd., consists of rubber blocks of suitable segmental shape inserted between the blades or projections of an inner rotor and the internal blades of an outer member which also serves as a casing (Fig. 45). The blocks, which may be of natural rubber or neoprene, and which are available in different degrees of hardness, thereby giving a wide choice of torsional stiffnesses, are totally enclosed. As the blocks are initially precompressed before being driven into the cavities formed by the driving and driven members,

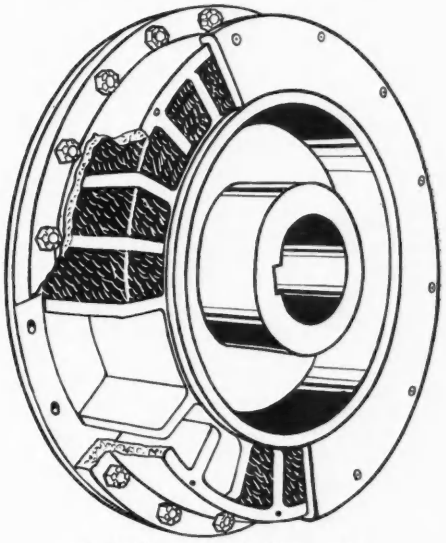


Fig. 45. **Heavy-duty rubber-block coupling.**
(*Courtesy of The Holset Engineering Co. Ltd.*)

complete unloading of the blocks, even under severe torque fluctuations, is prevented. Moreover, owing to their variable stiffness, they have a detuning action, which reduces to some extent the vibration amplitudes of the system in which they are used. These couplings, which are very easily dismantled and which accommodate parallel and angular misalignments, are available with maximum power ratings from about 180 hp at 6000 rpm for a 7½-in. coupling to 29,000 hp at 60 rpm for a 79-in. coupling. They can also be supplied with a special shear-pin device for protection against overloads, while, for diesel-engine applications, where a very high degree of resilience is required, they can be fitted with cylindrical rubber blocks, instead of the normal segmental blocks, so as to provide torsional deflections up to 6 deg. In addition, these couplings are incorporated in a range of Holset resilient gear-wheels of compact design.

A coupling of the " Neidhart " type, made in Great Britain by the André Rubber Co. Ltd., is illustrated in Fig. 46. In this type of unit, rubber cylinders are pressed between two metal bodies of hexagonal shape, the rubber units being unbonded to the metal and acting in rolling compression. One of the main features of this coupling is the large torsional deflection possible, i.e., 30 deg.

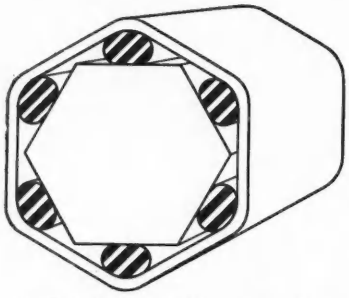


Fig. 46. André-Neidhart rubber-block coupling.
(Courtesy of André Rubber Co. Ltd.)

Units of this type have been produced with rubber cylinders ranging from $\frac{1}{8}$ to 4 in. in diameter, with respective capacities of from about 35 lb per inch length to about 35,000 lb/in. These couplings lend themselves to easy incorporation in machine elements, e.g., flywheels, while their construction is such that no machining whatsoever of the metal faces contacting the rubber is required.

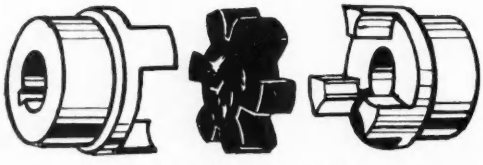


Fig. 47. Rubber spider coupling with six arms.
(Courtesy of Gerbing Manufacturing Corp.)

Rubber spider couplings are somewhat similar to dog-type couplings, but direct metal-to-metal contact is avoided in this case by interposing a rubber spider. An example of a rubber spider coupling, made in the U.S.A. by the Gerbing Manufacturing Corporation, is shown in Fig. 47, ratings being from $1\frac{1}{2}$ to 30 hp at 1750 rpm, according to size. In this particular design, the six-armed spider is made of specially compounded " Buna N " synthetic rubber, which is not only heat- and oil-resistant and stable over a wide range of temperatures, but is capable of withstanding continuous flexures and shock loads and of deforming to compensate for misalignment. Furthermore, the arms of the spider, as a result of the specially designed jaw faces, are held in position in such a way that any tendency of the arm to extrude and break is eliminated.

A spider coupling produced in England by Renold Chains Ltd. is shown in Fig. 48, and comprises two cast-iron or bronze members with concave-faced jaws fitting around a moulded-rubber four-armed spider. This compact design is suitable for high-speed operation, maximum ratings being from 1 hp at 10,000 rpm for a 1-in. coupling to 20 hp at 4000 rpm for a $3\frac{3}{4}$-in. coupling. The method of attachment of this unit, which can accommodate angular misalignments up to 2 deg., does not involve any projections, attachment to the shafts being

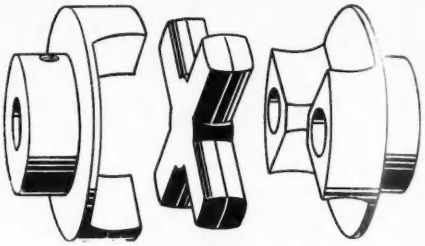


Fig. 48. Rubber spider coupling with four arms.
(Courtesy of Renold Chains Ltd.)

effected simply by means of set-screws in each half-body. Another British spider coupling of similar design, is the " Harrisflex ", made by Howard Clayton-Wright, in which the rubber spider is sometimes replaced by a nylon spider for increased torque capacity.

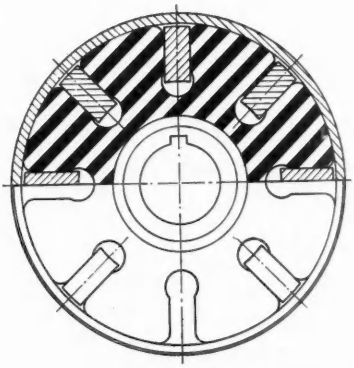


Fig. 49. Holset spider coupling.
(Courtesy of The Holset Engineering Co. Ltd.)

The Holset spider coupling (Fig. 49), although basically following the same principle as their heavy-duty rubber-block coupling (Fig. 45), utilizes rubber blocks moulded in one unit, i.e., in the form of a spider. This coupling is designed to eliminate bolting and to enable the coupling halves to be assembled easily and quickly by axial movement only. This feature makes it particularly useful for operation with flange-mounted motors. Ratings are from about 30 hp at 2000 rpm for a 6-in. coupling up to 500 hp at 1500 rpm for a 12-in. coupling.

Another type of rubber spider coupling is made in Italy under the Pirelli " Saga " patent, by Soc. Applicazioni Gomma Antivibrante. This coupling (known in the U.K. as " Jubo " (Fig. 50), and manufactured by The Avon India Rubber Co. Ltd.) uses four blocks which are made from a soft rubber compound and assembled in a single spider located in a forged aluminium-alloy housing. This type of coupling can accommodate

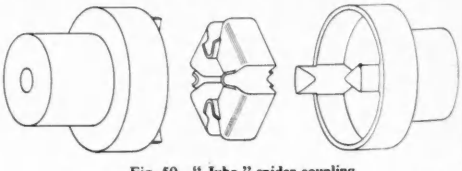


Fig. 50. " Jubo " spider coupling.
(Courtesy of The Avon India Rubber Co. Ltd.)

parallel, axial, and angular misalignments, and torsional deflections up to 15 deg. (normal torque) or 25 deg. (maximum instantaneous torque). Power ratings at 1500 rpm continuous operation range from 1½ hp for a 2¼-in. coupling to 12 hp for a 4-in. coupling. In France, a somewhat similar design, known as the " Miniflex " coupling (Fig. 51), is produced by Usines Paulstra.

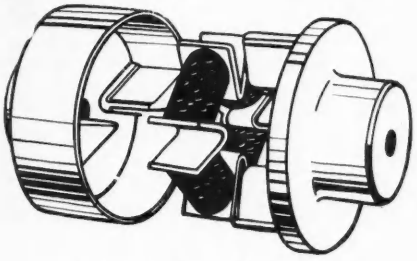


Fig. 51. " Miniflex " spider coupling.
(Courtesy of Usines Paulstra)

Included in this section is the " British-American " flexible coupling (Fig. 52), manufactured in the U.K. by R. Hunt & Co. Ltd. This coupling is not strictly a spider coupling, but is rather an improved application of the well-known Oldham principle, in which a floating centre member is constrained to slide across the face of one coupling flange on a line passing through its centre. At the same time, this centre member is free to slide across the face of a second flange in a direction at right angles to the first, thus combining flexibility with precision of movement. In the " British-American " coupling, the floating centre member is a square, hollow casting, to the edges of which are attached easily replaceable bearing strips made from a tough plastic, which has excellent wearing qualities and which resists the action of heat, moisture, and dirt. The hollow centre member is filled with a grease lubricant, and has four small holes on each bearing surface, the holes being equipped with wooden reeds leading to felt pads in the special bearing strips. Thus, when the coupling revolves, centrifugal force assists in providing perfect lubrication of the bearing strips. In normal operation, the centre member contains sufficient lubricant for 1000 hr, after which it can easily be refilled by removing one of the bearing-strip screws and injecting the grease with an ordinary grease gun. By virtue of the fully floating centre member, vibration is arrested, and the coupling, which allows for axial movement at full load, as well as for axial misalignment, does not set up cross pulls or end thrust. Maximum power ratings range from about 1½ hp at 10,000 rpm for a 2¼-in. coupling to 3400 hp at 1400 rpm for a 30-in. coupling.

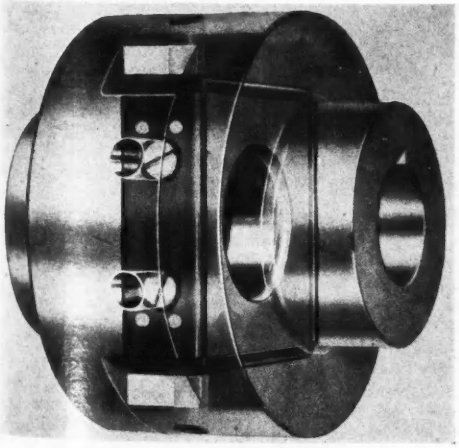


Fig. 52. " British-American " coupling.
(Courtesy of R. Hunt & Co. Ltd.)

Radial rubber-spring couplings are basically similar to rubber-block couplings, except that, in the radial spring arrangement, only the ends of the rubber blocks are located in peripheral recesses of outer and inner carrier rings of one coupling member and are acted upon by projections on the other coupling member. In rubber-block couplings, the blocks are mainly in compression, whereas, in radial spring couplings, the blocks are loaded in bending and in shear. An example of a radial spring coupling, known as the " Eupex " and made in Germany by Flender and under licence in England by Armstrong Whitworth (Metal Industries) Ltd., is shown in Fig. 53. Couplings of this type accommodate parallel, axial, and angular misalignments, and torsional deflections up to 6 deg., and are made for maximum power ratings from about 7 hp at 9500 rpm for a 3-in. coupling to about 27,000 hp at 335 rpm for an 80-in. coupling.

Fig. 53. " Eupex " radial rubber-spring coupling.
(Courtesy of A. Friedr. Flender & Co.)

Another design of radial rubber-spring coupling is made in Germany by Breitbach, in which rubber blocks are fitted in pairs on either side of stroke-limiting metal webs or arms which support the blocks on one side.

(e) *Flexible Ring or Link Couplings*

Two totally different types of coupling come under this heading, i.e., (1) flexible ring-like couplings with holes for pins of the driving and driven members, these being akin to disc couplings, and (2) couplings consisting of plates with a number of studs or projections linked together by rings of leather, rubber, or other materials. The " Rotoflex " coupling, produced in the U.K. by Metalastik Ltd., is a flexible-ring coupling of the first type, and is shown in Fig. 54. In this design, the rubber ring consists of a number of solid-rubber cylindrical sections bonded together, steel bushes or tubes passing through

and enabling the coupling to be bolted to the driving and driven flanges. An additional centering device is incorporated in these couplings when used for high-speed operation. These couplings, which provide considerable torsional flexibility, and which accommodate angular misalignment and considerable end-float, are particularly suitable for cardan-shaft assemblies. " Rotoflex " couplings were developed and are manufactured in Italy by Soc. Applicazioni Gomma Antivibrante ; in addition to Metalastik Ltd., they are also made by other manufacturers, including Usines Paulstra in France and Belgium, typical power ratings at 100 rpm being about 5 hp for a 4½-in. coupling up to about 40 hp for an 11½-in. coupling.

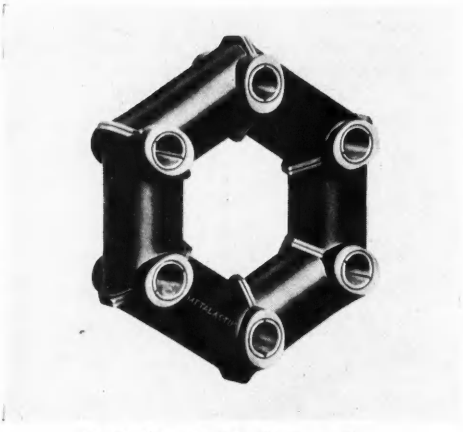


Fig. 54. " Rotoflex " flexible-ring coupling.
(Courtesy of Metalastik Ltd.)

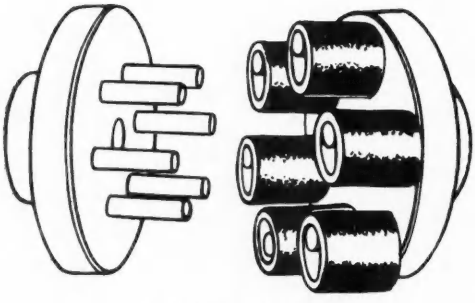


Fig. 55. " Rex " link coupling.
(Courtesy of Gebr. Rykart)

An example of a coupling of the second type is shown in Fig. 55. Known as the " Rex ", this coupling, which is made in Switzerland by Gebr. Rykart, uses two flanged hubs with pins on different pitch-circle diameters, coupled together by rubber rings. Reduction of shock, reversibility, and accommodation of axial float and misalignment are properties that can be expected of this type of coupling, which is supplied in a range of sizes for powers from about 16 hp at 8000 rpm for a 3-in. coupling, to 1650 hp at 1000 rpm for a 26-in. coupling.

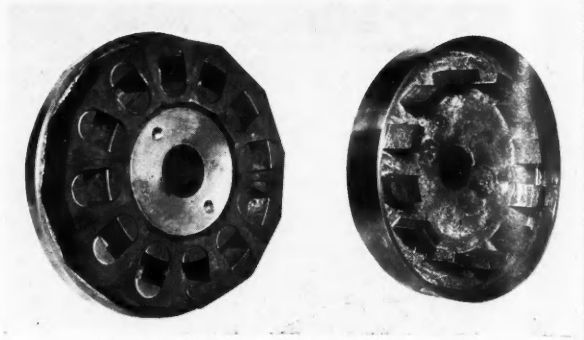


Fig. 56. Whittaker Hall elliptical rubber-buffer coupling.
(Courtesy of Whittaker Hall & Co. (1929) Ltd.)

The coupling shown in Fig. 56, produced in England by Whittaker Hall, comprises one coupling half with axial metal projections or lugs which engage with a corresponding series of rubber rings, designed as buffers with elliptical recesses, carried on pairs of lugs on the other coupling half. The shape and flexibility of the rubber buffers ensure that even pressure is exerted by each driving lug. The external finish is absolutely smooth and free from projecting parts. These couplings, which are suitable for either direction of rotation, and which can accommodate parallel, axial, and angular misalignments, are available for power ratings ranging from about 18 hp at 12,000 rpm for a 2⅜-in. coupling up to 2000 hp at 1350 rpm for a 24-in. coupling.

(*f*) *Belt and Rope Couplings*

Although belt and rope couplings may nowadays be considered virtually obsolete, some particulars of these couplings are included in this survey for the sake of completeness.

A typical belt coupling comprises inner and outer halves of cast iron or steel, connected by a tough flexible belt which passes over and under a number of peripheral lugs on the driving and driven members of the coupling, the number and width of the loops so formed being determined by size and power requirements. If reversibility is required, two separate belts can be employed, one for each direction of rotation. Couplings of this type, which absorb shock and can accommodate slight misalignments, are particularly applicable for high powers at low speeds, e.g., 2000 hp at 200 rpm for a 72-in. non-reversible coupling, but are not suitable for use in damp, hot, or very dry atmospheres.

The rope-type coupling generally comprises two cast-iron flanged discs with studs secured in each half at different radii from the centre, these studs being spaced alternately. A manilla or cotton rope is wound over and under alternate studs and, by removing the rope, either shaft can rotate independently, while, by removing the studs, each shaft can be lifted clear. Couplings of this type, which can absorb shock and can accommodate misalignments, are suitable for general industrial drives, particularly in isolated districts, as manilla or cotton rope is usually obtainable everywhere. A typical maximum power rating is about 650 hp at 750 rpm for a 27-in. coupling.

ALL-METAL COUPLINGS

To a considerable degree, all-metal flexible couplings have properties which are very similar to those of non-metallic flexible couplings ; however, they tend to predominate mainly in applications where high or very high torques are to be transmitted, where high or very high operating speeds are required, and where very high or very low operating temperatures are expected. The range of flexibility attainable with all-metal couplings is, in certain designs, at least as great as that obtained with non-metallic couplings and, in addition, there is a larger number of metallic designs which provide for some detuning action and for shock-load absorption as an inherent feature, where a certain amount of slackness is available in construction. However, particularly with very high-speed designs, all-metal couplings usually offer narrower limits for misalignment, although in medium- and low-speed applications the limits of permissible movement are at least equal to those of non-metallic couplings.

(a) All-Metal Disc Couplings

In its simplest form, a flexible all-metal disc coupling comprises a flexible steel disc or discs which, near their periphery and around a pitch circle of smaller diameter, are bolted to corresponding flanges on the driving and driven shafts. The main virtue of this type of coupling is that it permits some axial movement and slight angular misalignment, while still being capable of carrying considerable torques. In single- or double-disc systems, large angular errors superimpose appreciable bending stresses in the disc, and these cannot be tolerated ; furthermore, no accommodation of radial misalignment is provided in the simplest systems.

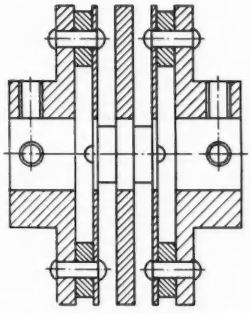


Fig. 57. **All-metal disc coupling with three intermediate discs.** (*Courtesy of Oerlikon Tool & Arms Corp. of America.*)

Since rubber or fabric discs give greater flexibility, it may well be asked when all-metal discs should be used. The answer is that all-metal discs can operate under all kinds of extreme conditions of temperature, dust, etc. which may not be favourable in the case of rubber or fabric disc couplings. Moreover, in some cases, from the point of view of torsional-vibration frequencies, it is preferable to utilize a " close-coupled " system, so that in those cases where the requirement is for a fairly stiff drive which will, however, also accommodate some axial or slight angular misalignments, steel discs are employed. Steel discs have a high torsional stiffness ; in fact, the torsional flexibility of a steel disc coupling is mainly due to the deflections in its bolting arrangements. Another specific feature is that, since steel disc couplings have no rubbing or mating elements, they can truly be said to have no wearing parts ; in addition, they are easily disconnected, merely by removing the discs, and are therefore sometimes preferred to other arrangements involving the fitting and removal of keys.

For increased ability to accommodate end-float, groups of steel discs can be mounted together. In one design, the Series C-300 coupling (Fig. 57), made by the Oerlikon Tool & Arms Corporation of America, three intermediate discs, riveted between two flanged hubs, are used. This coupling can accommodate angular misalignments of up to 6 deg.

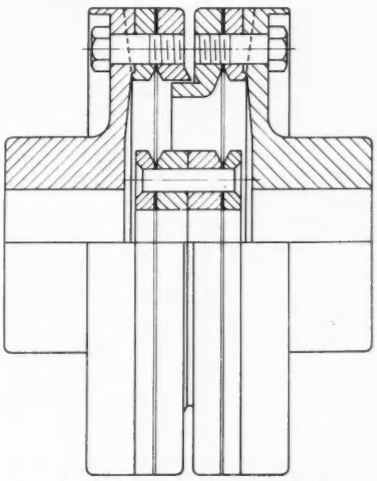


Fig. 58. **" Metastream " metal disc coupling.** (*Courtesy of Metaducts Ltd.*)

The " Metastream " coupling, produced in Great Britain by Metaducts Ltd., is illustrated in Fig. 58. In this all-metal non-lubricated coupling, the drive is achieved through flexible discs which are made up of a number of stainless-steel laminations, with from four to eight spokes, depending on size (Fig. 59). Torque is transmitted in shear, the coupling being basically rigid, and resistance to axial movement is controlled by the number of flexible lamina-

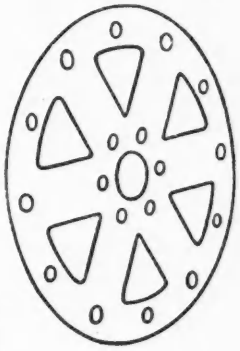


Fig. 59. **Lamination with six spokes for " Metastream " metal disc coupling.** (*Courtesy of Metaducts Ltd.*)

tions used in each disc, this number being varied to suit individual requirements. To ensure maximum flexibility, two discs are used in each coupling. There is no angular backlash or play between driving and driven units and, under all conditions of load, a positive drive and perfect balance are ensured, whether the coupling halves are in alignment or otherwise. The axial flexibility of the coupling is unaffected when it is transmitting its full rated torque, and it will operate satisfactorily under conditions of $1\frac{1}{2}$-deg. angular misalignment, combined with a reasonable amount of parallel misalignment. In addition, the flexible-disc unit can be removed and re-assembled very quickly, without disturbing the driving or driven units. A typical coupling, partly cut away to show construction, is illustrated in Fig. 60.

Fig. 60. Cut-away view of " Metastream " metal disc coupling. *(Courtesy of Metaducts Ltd.)*

" Metastream " couplings are normally made in sizes capable of transmitting from 30 hp at 15,000 rpm for a $2\frac{7}{8}$-in. coupling up to 8000 hp at 1000 rpm for a 26-in. coupling, but special couplings have been supplied for extra-high-speed duties (e.g., 200 hp at 30,000 rpm) and for prototype marine main propulsion machinery up to 15,000 hp at 6000 rpm. To permit some parallel misalignment and to facilitate disassembly, " Metastream " couplings are also supplied complete as spacer types, with a pair of couplings used in conjunction with a short floating jackshaft.

When only a limited amount of axial space is available, the spacer can be replaced by a floating centre-plate, as in a design produced in the U.S.A. by the Thomas Flexible Coupling Co. This design utilizes annular discs to avoid bending stresses, so that it may be termed a flexible-annulus type of coupling. Somewhat similar couplings, but of smaller size for servo-mechanism applications, are also made in the U.S.A. by Renbrandt, Inc. These miniature couplings, some of which are as small as $\frac{1}{2}$ in. in diameter by $\frac{11}{16}$ in. in length, have zero backlash and low inertia, and do not introduce velocity variations between the driving and driven shafts.

A different type of disc coupling has been produced in the U.S.A. by the York Corporation to drive their centrifugal compressors. In this design, the flexible steel discs are made integral with the hubs, which are threaded to the

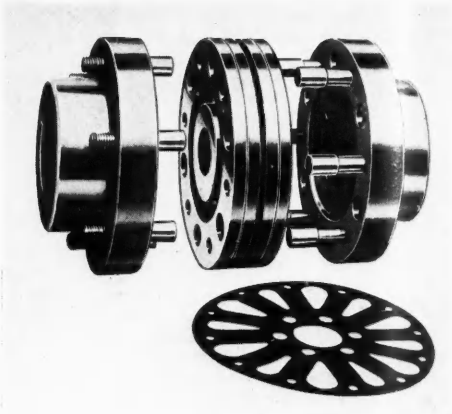


Fig. 61. " Uniflex " metal disc coupling. *(Courtesy of Crofts (Engineers) Ltd.)*

shafts. The discs are located in recesses between a comparatively thick aluminium-alloy spacer, the outer clamping rings also being of aluminium alloy. By connecting this coupling to the compressor shaft via a flexible rod, the transmission of torque pulsations is reduced and local fatigue failures in the rotor are prevented.

The " Uniflex " metal disc coupling (Fig. 61), made in the U.K. by Crofts (Engineers) Ltd., incorporates two flexible metal discs of stainless steel or monel metal, with triangular slots, which are clamped between outer and central plates, the outer plates being mounted on steel pins fitted on the flanged hubs. The pins are extended into free holes in the opposing half coupling, ensuring continuity of drive in the event of failure under excessive shock loads. No lubrication is required and the sandwich type of construction used permits easy fitting or removal of the transmission unit without disturbing either the driving or the driven half of the coupling. A further feature is its complete freedom from backlash, even under extreme loading conditions. These couplings accommodate axial, angular, and some parallel misalignment, and have maximum power ratings ranging from about 40 hp at 12,850 rpm for a $3\frac{1}{2}$-in. coupling to 8800 hp at 1960 rpm for a $23\frac{3}{8}$-in. coupling.

Flexible all-metal disc couplings, specially constructed to suit individual requirements, are also made in the U.K. by Standage Power Couplings Ltd, who specialize in the design and construction of a wide range of types of couplings for special applications.

(b) Metal Spring Couplings

Under this heading, such a large and important group of couplings is included that it is desirable to subdivide metal spring couplings into different classifications, i.e., (i) helical spring couplings, (ii) radial spring couplings, (iii) peripheral blade-spring and spring-grid couplings, and (iv) sleeve spring couplings.

(i) Helical Spring Couplings:—

The simplest form of helical spring coupling is shown in Fig. 62, in which the ends of a helical spring are inserted into radial holes in the end portions of the driving and driven shafts. In this way, a considerable amount of axial misalignment is obtainable, at the cost of some end

thrust, and the torque is transmitted by the spring. This type of coupling also accommodates parallel and angular misalignments. In somewhat different designs, the spring ends can be inserted into bosses on the shafts and secured in position by means of grub-screws or by clamping arrangements. Torque and speed limits are comparatively low, so that couplings of this particular type are mainly suitable for fractional-horsepower and low-speed applications. Preferably, they should be used in only one direction of rotation, i.e., the direction in which the torque tends to tighten the coils of the spring.

Fig. 62. **Simplest arangement of helical spring coupling.**

An example of a circumferential helical spring coupling is illustrated schematically in Fig. 63, in which a six-bladed inner member acts on springs located between the internal blades of the outer or driven member. The springs are mounted in the axial direction by means of a plate with peripheral slots, through which parts of the sides of the coil springs can protrude. Couplings of this type are sometimes made by internal combustion engine manufacturers for particular applications, and can accommodate parallel, axial, and some angular misalignments. As the " rate " or linear stiffness of the helical springs is constant, these couplings have a constant torsional stiffness and also a comparatively high torque-carrying capacity. Owing to some inherent slack in the mechanical arrangement, they can absorb impacts to some extent. To reduce the effects of torque reversals, the spring elements are usually precompressed before insertion into their recesses.

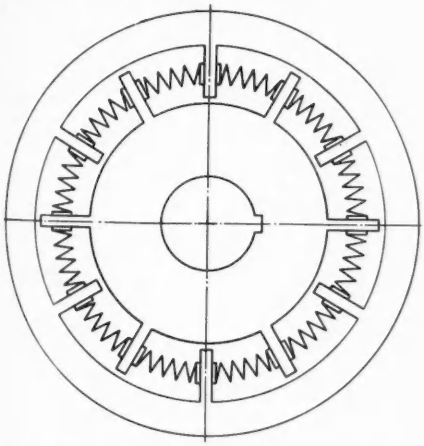


Fig. 63. **Schematic arrangement of a circumferential helical-spring coupling.**

A helical spring coupling for main or auxiliary drives, particularly for marine applications, is shown in Fig. 64. Couplings of this type, which are made in the U.K. by Modern Wheel Drive Ltd., are capable of transmitting powers in excess of 2000 hp. Designated the " Hindmarch/MWD " flexible coupling, these units have been developed for use between a diesel engine and a reduction or reverse reduction gear, and are specifically designed to exercise a detuning action, in that stiffness variations occur in such a manner that the torsional " critical " speeds of the driving and driven systems can be eliminated from the speed range. As shown in Fig. 64, this coupling is of the peripheral helical-spring type but, in addition, it incorporates " buffer " springs, located over the normal inner springs, giving a rapid increase in stiffness after the

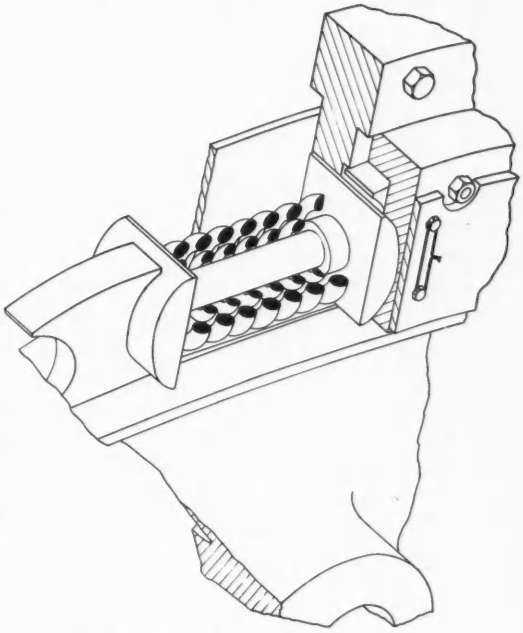


Fig. 64. **" Hindmarch/MWD " helical spring coupling.**
(Courtesy of Modern Wheel Drive Ltd.)

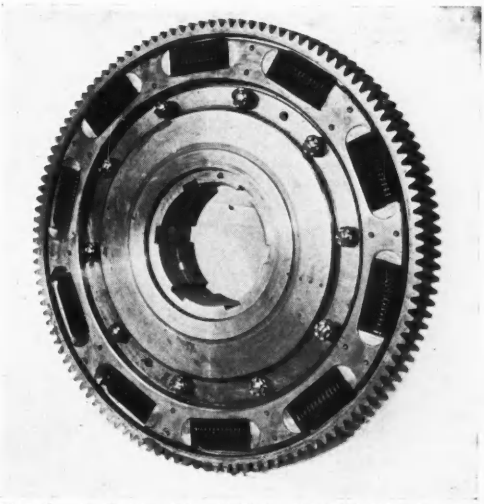


Fig. 65. **Application of the " Hindmarch/MWD " helical spring coupling to a flexibly mounted gear-wheel.**
(Courtesy of Modern Wheel Drive Ltd.)

maximum mean torque has been exceeded. In this way, any vibratory torques above the limiting value will raise the coupling stiffness, and also, therefore, the vibration frequency of the system in which it is interposed, so that critical speeds will be shifted above the operating range. Although these couplings are not primarily intended to accommodate misalignments, typical amounts are 0·002 in. parallel misalignment, 0·1 in. axial misalignment, and 10 minutes angular misalignment. The torsional stiffness of these couplings can be positively varied over a wide range up to an equivalent of approximately 10 deg. torsional deflection. A certain amount of damping is also obtained, but their detuning action is the predominant feature. An application of the Hindmarch/MWD coupling to a flexibly mounted gear-wheel is shown in Fig. 65.

(ii) Radial Spring and Modified Oldham Couplings:—

In some designs of radial spring coupling, e.g., that shown schematically in Fig. 66a, blade springs are clamped at both ends, i.e., they are attached at one end to the inner member, and at the other end to the outer member. Somewhat greater flexibility is obtained with the design shown in Fig. 66b, in which the blade springs are clamped at the periphery of the inner member only and are provided with pivots in semicircular recesses in the periphery of the outer member. Still more flexibility is obtained by having both ends of the spring free (Fig. 66c). In this design, packs of loosely fitted steel springs reduce shock loads, owing to their slackness. Radial spring couplings, with either one or both ends of the springs free, can accommodate appreciable angular, parallel, and axial misalignments.

A radial spring coupling, specially designed for connecting standard motor shafts to abnormally large flanges, flywheels, rolls, etc., is shown in Fig. 67. This coupling, which is manufactured in the U.K. by Standage Power Couplings Ltd., is of overhanging design and combines economy of space with the ability to transmit very high powers (depending on the width of the springs used). The maximum power rating of the 14¾-in. coupling illustrated is about 40 hp per 100 rpm, but greater powers can be transmitted by increasing the width of the springs, without increasing the overall diameter of the coupling.

A radial spring coupling which is, in effect, a modified Oldham type of coupling, is also manufactured by Standage Couplings Ltd. As can be seen from Fig. 68, this design basically comprises a floating disc with four sets of cast-in spring packets which engage in recesses in two identical hub members. With this design, the advantages of the Oldham principle in accommodating extreme parallel misalignment are combined with the ability to absorb torsional shocks and vibration, and a certain amount of angular misalignment. Maximum power ratings of this coupling range from about 40 hp per 1000 rpm for a 4½-in. coupling up to about 300 hp per

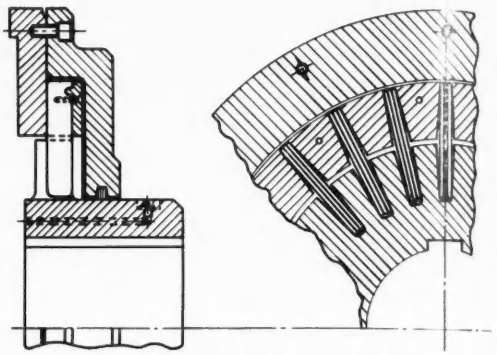


Fig. 67. **Special overhanging design of radial spring coupling.** (*Courtesy of Standage Power Couplings Ltd.*)

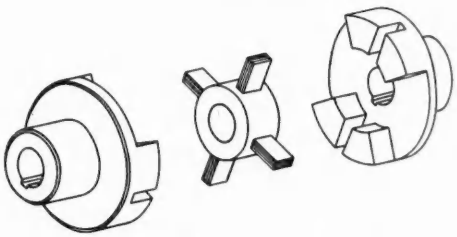


Fig. 68. **Modified Oldham-type of radial spring coupling.** (*Courtesy of Standage Power Couplings Ltd.*)

1000 rpm for a 10½-in. coupling. By increasing the length in shear of each spring and the number of laminations per pack, larger powers can be transmitted. The design shown in Fig. 68, in which the springs radiate from an internal central member, is intended for light to medium duties only. For heavy-duty applications, or where no gap between the shaft ends can be tolerated, the springs are held at the periphery of the coupling and radiate inwards.

(iii) Peripheral Blade-Spring and Spring-Grid Couplings:—

An example of a peripheral blade-spring coupling is illustrated in Fig. 69, this being the "Steel-Shaw" coupling, made in England by Steele & Cowlishaw Ltd. This design comprises two cast-steel hubs, keyed to their respective shafts, slots or grooves being cut in the periphery of the hubs. Separate floating steel keys or springs are located in the slots, the springs being so shaped that their ends are a close sliding fit in the slots and having a long free span between the points of support. The springs

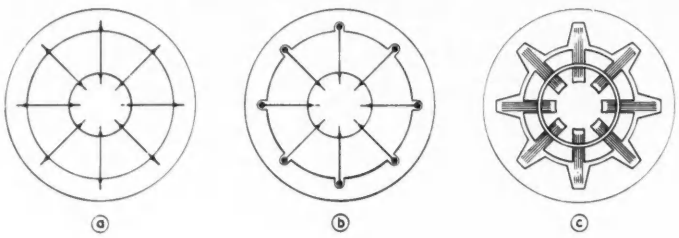


Fig. 66. **Schematic designs of radial spring couplings, showing arrangement: (a) with blade springs clamped at both ends, (b) with blade springs clamped to the periphery of the inner member only, and (c) with both ends of the blade springs free.**

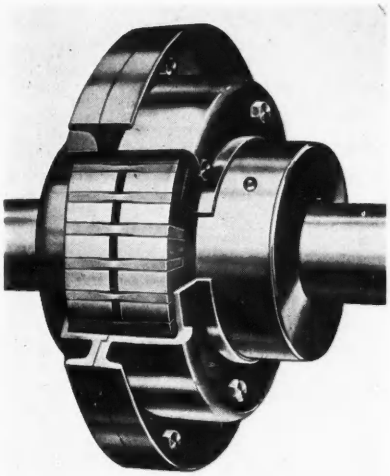


Fig. 69. **" Steel-Shaw " peripheral blade-spring coupling.**
(Courtesy of Steele & Cowlishaw Ltd.)

are waisted towards their middle length, the sides being shaped in a parabolic curve, so that, as load increases, so also does the area of contact between the slots and the spring, thus automatically decreasing the span and supporting each spring against bending. Flexibility is retained, even when serious misalignment occurs and under extreme emergency conditions. The coupling instantly accommodates all errors in alignment, absorbing shocks and stresses, damping out vibrations, and allowing free end-float under loads. A lubricant is packed within the coupling itself, so that the sliding contacts of the springs in their slots are continuously lubricated, eliminating wear of load-carrying surfaces. Furthermore, the coupling is self-contained and dustproof, and disconnection of the shafts, without end movement, is easy and rapid. Maximum power ratings range from about 60 hp at 4860 rpm for a 4¼-in. coupling to about 13,000 hp at 530 rpm for a 38-in. coupling.

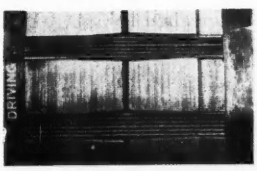


Fig. 70. **Peripheral blade-spring coupling with laminated springs.**
(Courtesy of R. Hunt & Co. Ltd.)

A range of peripheral blade-spring couplings of somewhat similar design is manufactured in the U.K. by R. Hunt & Co. Ltd. In this design, however, laminated springs are used and, instead of waisting the springs in the middle, the middle portions of the slots are flared, i.e., there is normally a Vee-shaped gap between the springs and the slot walls, as shown in Fig. 70. The maximum power ratings of these couplings range from about 18 hp at 6000 rpm for a 3¾-in. coupling to about 7100 hp at 940 rpm for a 24¼-in. coupling.

Peripheral blade-spring couplings are also produced in the U.K. by Standage Power Couplings Ltd. These couplings, which also use laminated springs and flared slots, are available with cast-iron or steel covers, maximum power ratings (cast iron) ranging from about 26 hp at 5300 rpm for a 4½-in. coupling to 10,000 hp at 830 rpm for a 30-in. coupling. For couplings with steel covers, these speeds can be doubled. In addition, special high-speed types with pressed steel covers, extensively used on diesel railcars, turbo-generators, etc., are available, together with self-supporting types, developed for those drives in which only one bearing is fitted near the coupling. All these couplings are capable of accommodating up to 3 deg. angular misalignment and 0·020 in. parallel misalignment simultaneously.

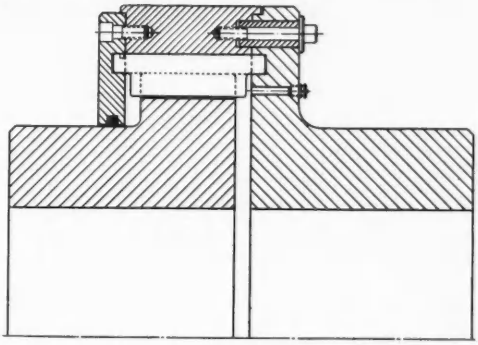


Fig. 71. **Special design of blade-spring coupling using a combination of bowed and flat springs, as shown on left.**
(Courtesy of Standage Power Couplings Ltd.)

A particularly interesting development by Standage Power Couplings Ltd. is the coupling shown in Fig. 71. In this design, the two halves are not coupled by the springs across their periphery, but are arranged concentrically, so that the springs engage the inner and outer members radially. This coupling is therefore, in effect, a radial coupling, the great advantage of this particular design being that the power to be transmitted is controlled by the length of the springs, so that, by flexing across their length instead of their breadth, the same springs can be made to transmit considerably higher powers, without an increase in coupling diameter. Thus, for example, a peripheral blade-spring coupling of this firm would be about 56 in. in diameter for an application involving the transmission of 6000 hp per 100 rpm, whereas this radial coupling would have a diameter of only 29½ in. for the same power rating. Another feature of this coupling is the fact that, as shown in Fig. 71, bowed springs, or a combination of bowed and flat springs, can be used, to produce any desired detuning action, e.g., in diesel-engine applications. The maximum power ratings of these couplings are from about 2150 hp at 3600 rpm for a 6¾-in. coupling up to any desired value, e.g., 30,000 hp at 830 rpm for a 36-in. coupling.

A spring-grid coupling incorporating independent C-shaped springs is shown in Fig. 72, which illustrates the " Multiflex " coupling, made in the U.K. by Crofts (Engineers) Ltd. In this design, the grooves on one hub are straight with slightly rounded ends, whereas on the other they have an inwardly flared or slightly tapered contour, the C-springs being inserted into the grooves with their " free " ends between the flared sections.

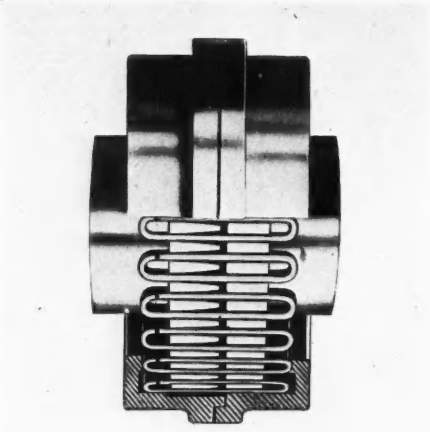


Fig. 72. " Multiflex " spring-grid coupling with C-shaped springs. (*Courtesy of Crofts (Engineers) Ltd.*)

Owing to the slight inherent slack in the assembly, this coupling is capable of reducing shock loads and, through its detuning action, of reducing torsional-vibration amplitudes. Stiffness can be adjusted to obtain an optimum value by removing diametrically opposite independent springs and replacing them by other sizes of spring, if necessary. This coupling, which is comparatively flexible and has a high torque-carrying capacity, has maximum power ratings from about 70 hp at 4750 rpm for a $4\frac{3}{8}$-in. coupling to 15,000 hp at 300 rpm for a 60-in. coupling. Typical applications of these couplings, which are fitted with oiltight, dustproof covers totally enclosing the springs, include rolling mills, pumps, haulage gears, and turbines.

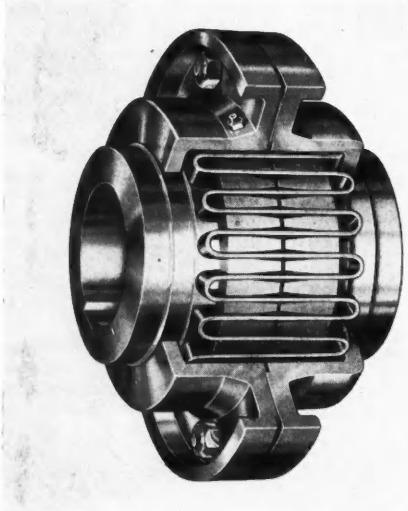


Fig. 73. " Bibby " spring-grid coupling. (*Courtesy of The Wellman Bibby Co. Ltd.*)

A " Bibby " spring-grid coupling is shown in Fig. 73. This type of coupling, first designed and constructed in the U.K. by The Wellman Bibby Co. Limited, is also manufactured in the U.S.A. by The Falk Corporation (under the name of the " Steelflex " coupling), in France by Établissements Citroën, and in Germany by Maschinenfabrik Geislingen & Co. Available in a wide range of types and sizes, the Bibby coupling is used in horizontal and vertical installations in almost every conceivable type of drive in stationary, marine, and aircraft applications. Originally developed nearly forty years ago (Bibby couplings were fitted to the Maybach engines in the airship " Graf Zeppelin "), it still remains one of the most effective detuning couplings in existence.

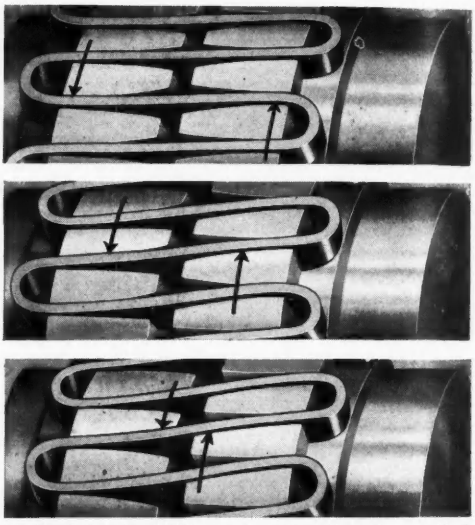


Fig. 74. Action of the " Bibby " spring-grid coupling under normal load (*top*), normal overload (*middle*), and severe accidental overload (*bottom*). (*Courtesy of The Wellman Bibby Co. Ltd.*)

This highly resilient coupling consists of two thick discs with axial grooves around their peripheries, connected by one or more layers of spring grids. The grooves are flared towards the middle of the blade sections of the grids, and the peripheries of the coupling halves can therefore be regarded as having a large number of curved teeth, along which the blade-spring portions are supported to an extent which varies according to the amounts of steady and vibratory torques transmitted to the coupling at any particular instant. Since the stiffness of the blades (and of the coupling) mainly depends on the free or unsupported length of the blade-spring portions, and since this varies with every variation in sinusoidal torque, coupling stiffness changes at every instant during a vibration cycle, and thus a powerful detuning action is produced, which continuously alters the torsional-vibration frequency of the system in which the coupling is incorporated, preventing a build-up of resonant amplitudes. The characteristics of the springs must be designed to suit the particular system under consideration, and there is abundant evidence to show that, under suitable conditions, torsional-vibration stresses are considerably reduced. The increased stiffness due to the shortening of the free lengths of the blade springs under load also enables the coupling to sustain high torques without overstressing the springs (Fig. 74). The discs or hubs forming the two halves of the Bibby coupling are keyed, shrunk, or flanged to the driving and driven shafts and are

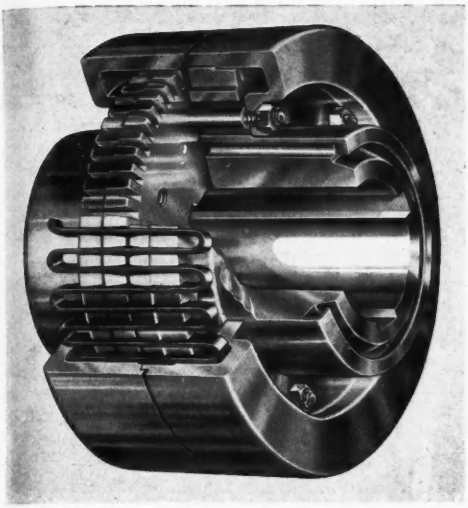


Fig. 75. **Special design of " Bibby " spring-grid coupling for turbines.** (*Courtesy of The Wellman Bibby Co. Ltd.*)

completely enclosed. In most cases, the coupling is grease-filled, although oil lubrication is used in high-speed applications.

A special design of Bibby coupling is provided for turbines (Fig. 75). For rolling-mill drives, the spring grids are of special cross-section, somewhat resembling an oval with flat sides, with two, three, or even four layers of grids located one over the other. In these and other applications, the coupling also serves to reduce low-frequency vibrations. The driving and driven shafts can easily be disconnected by removal of the spring grid.

Owing to the inherent resilience of the grid, the Bibby coupling reduces shock loads, often to one-half or one-third of the amount which would be experienced with a rigid coupling. It also accommodates a certain amount of parallel and angular misalignment and some axial movement, without introducing any appreciable axial or thrust forces. Couplings in operation are transmitting from 0·1 to 17,500 normal horsepower per 100 rpm, with peaks up to 50,000 hp per 100 rpm and speeds up to 20,000 rpm (or even higher for extra-high-speed designs). The outer diameters of these couplings range from 3¼ in. to 12 ft.

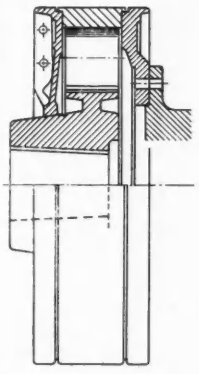


Fig. 77. **Renk sleeve spring coupling.** (*Courtesy of Zahnräderfabrik Renk.*)

(iv) Sleeve Spring Couplings:—

A representative example of an all-metal sleeve spring coupling is the " Deli ", shown schematically in Fig. 76. In this design, which is made in Germany by DEMAG, the inner rotor, keyed to the driven shaft, is provided with arc-shaped recesses round its periphery, the outer member being similarly constructed with internal recesses of correspondingly larger radius, and the packs of springs, consisting of helically slotted steel sleeves fitting into each other, are mounted between these two members. This

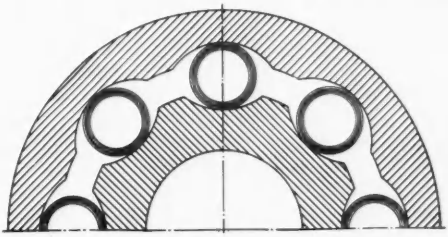


Fig. 76. **Schematic representation of the " Deli " sleeve spring coupling.** (*Courtesy of DEMAG.*)

coupling is extremely flexible (torsional deflections up to 10 deg. for certain sizes are quoted), and also accommodates angular misalignments up to 1 deg. An interesting feature of this design is that the sleeves presumably also act to some extent as rollers.

Another example of a sleeve spring coupling is made in Germany by Zahnräderfabrik Renk. This coupling (Fig. 77), which has a high torque-carrying capacity and which provides considerable detuning action, is used in marine propulsion systems, as well as in other applications. The outer or driving member has a number of internal semicircular recesses, which correspond to similar recesses in the inner or driven member, so as to form cylindrical cavities, into which the sleeve springs are inserted. Each pack of springs consists of slotted sleeves of gradually decreasing thickness, with the outermost sleeve having the greatest thickness, so as to have equal stress in all sleeves, and each pack is positioned by means of a centering bolt, preventing any relative slip between the sleeves. Variable stiffness (causing the detuning action which reduces vibration amplitudes) is obtained as a result of the shortening of the effective or " free " length of the sleeve springs with increasing deflection. The surfaces of the driving and driven members in contact with the packs of springs are suitably rounded and flared to avoid edge effects, and all springs are provided with means of lubrication. The fatigue strength in bending of these springs is stated by the makers to be about 1·7 times higher than the fatigue strength of torsion springs.

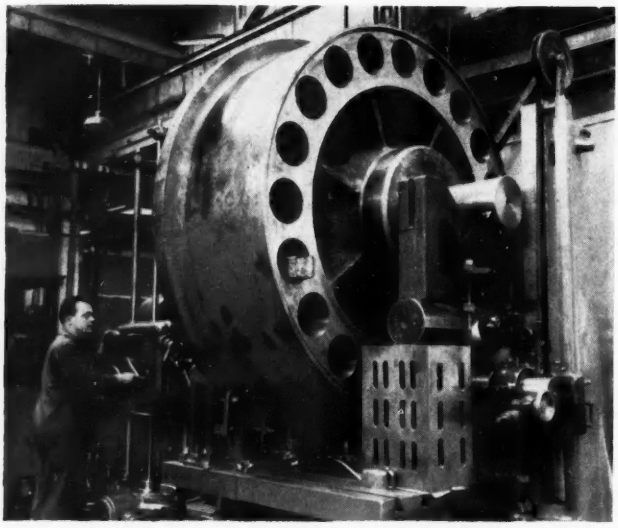


Fig. 78. **Example of a large Renk sleeve spring coupling.**
(*Courtesy of Zahnräderfabrik Renk.*)

Couplings of this type, which are suitable for absorbing shock loads, owing to the slack or play in their various components, mainly accommodate axial and parallel misalignments, as well as some angular misalignment. Torsional stiffness, depending on coupling size, number of springs, spring rate, and dimensions of the centering bolt, vary from about $1{\cdot}5 \times 10^5$ to $1{\cdot}5 \times 10^8$ lb-in./rad., and maximum power ratings range from about 50 hp at 3850 rpm for a $7\frac{1}{2}$-in. coupling up to about 20,000 hp at 560 rpm for a 52-in. coupling. Larger couplings, e.g., that illustrated in Fig. 78, are also available.

(c) *All-Metal Bush and Pin Couplings*

Under this heading can be included the " Flexpin " coupling (Fig. 79), manufactured in the U.S.A. by Smith & Serrell, Inc. This design basically consists of two hub members with a number of packs of steel springs suitably located in axial recesses around the hub periphery, bearing strips being welded to the outside layers of the laminations and inserted into self-lubricated bronze bushes in one coupling member. The laminations in the other coupling member are fitted into holders which are held in position by cross pins, permitting some pivoting action. Axial sliding motion of the steel laminations, which are copper-coated to resist rust, is free to occur in the bronze bushes, and these couplings, besides accommodating misalignments, reduce shock effects, owing to the slight amount of play in the assembly. Maximum power ratings range from about 25 hp at 4000 rpm for a $3\frac{3}{4}$-in. coupling up to about 20,000 hp at 490 rpm for a 37-in. coupling.

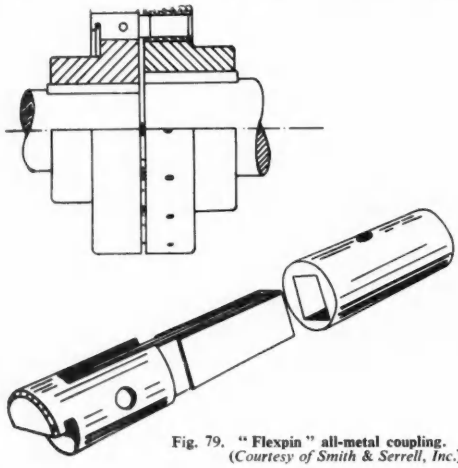


Fig. 79. **" Flexpin " all-metal coupling.**
(*Courtesy of Smith & Serrell, Inc.*)

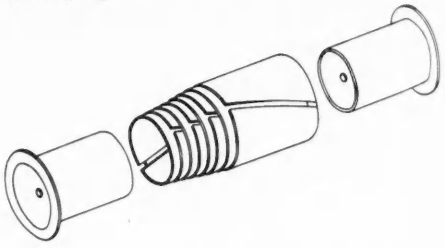


Fig. 80. **All-metal bush, intended to replace conventional rubber bushes in bush and pin couplings.**
(*Courtesy of Standage Power Couplings Ltd.*)

An interesting design of all-metal bush, intended to replace conventional rubber bushes in bush and pin couplings, is shown in Fig. 80. These bushes, which have been recently developed in the U.K. by Standage Power Couplings Ltd., basically comprise a number of accurately rolled slotted spring-steel members, nesting inside each other and mounted on brass sleeves. It is claimed that these metal bushes not only permit considerable misalignment without fatigue but are unaffected by climatic conditions, and provide greater durability and even more resilience than conventional rubber bushes.

(d) *Gear Couplings*

The principal characteristics of gear couplings (also known as spline or spindle couplings) are their compactness, associated with high torque-carrying capacity and negligible flywheel effect, their ability to operate at extremely high speeds, and their insensitivity to temperature within appreciable limits. In this large class of couplings, manufactured all over the world and deservedly popular, it is usual to distinguish between single-engagement and double-engagement couplings, the former having one set of inner and outer teeth, and the latter two such sets, coupled together by a floating member.

Originally, all gear couplings incorporated straight teeth or splines on both the hub and the housing. As a result, the capacity for angular and parallel misalignments was small, being controlled strictly by the amount of tooth clearance between the ring and the hub. Attempts to improve their capacity to accommodate such misalignments have resulted in the successful development of teeth with curved contours, i.e., crowned and barrelled, as shown in Fig. 81, and many manufacturers incorporate this feature.

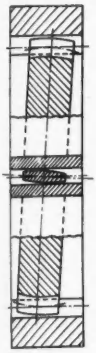


Fig. 81. Sectional view showing the function of crowned and barrelled teeth in gear couplings. *(Courtesy of F. Tacke Maschinenfabrik KG.)*

An example of a flanged-type double-engagement gear coupling with crowned and barrelled teeth is shown in Fig. 82. In this design, made in England by David Brown, the hubs are keyed to the driving and driven shafts, and the casing, consisting of two halves bolted together, is oil-filled, lubrication being ensured by centrifugal force. Couplings of this type intended for use in turbine drives are fed with a continuous supply of oil, either from a lip on adjacent bearings or from a small oil pipe, a drainage arrangement being in this case provided in the casing. Maximum power ratings are from 1 hp per 100 rpm for a $4\frac{1}{8}$-in. coupling up to 13,000 hp per 100 rpm for a 49-in. coupling.

A similar type of flanged double-engagement gear coupling with crowned and barrelled teeth is the " Bogenzahn ", manufactured in Germany by Tacke. These couplings, in addition to angular and axial misalignments, accommodate parallel misalignment from about $\frac{1}{16}$ to $\frac{3}{4}$ in., depending on type and on power and speed ratings, which range from about 1400 hp at 20,000 rpm for a $5\frac{3}{4}$-in. coupling to 193,000 hp at 550 rpm for a 70-in. coupling.

The " Sphereflex " coupling, produced in the U.S.A. by the Philadelphia Gear Works Inc., is a flanged-type double-engagement gear coupling employing teeth with curved roots and spherical flanks. This design is stated to compensate for shaft misalignments up to 14 deg. included angle. Another American gear coupling with crowned and barrelled teeth is made by the Poole Foundry & Machine Company. A variation of this principle is found in the gear couplings made in the U.S.A. by the Ajax Flexible Coupling Co. Inc., in which the teeth do not have a curved crown but incorporate dihedral, i.e., their contour is approximately in the shape of a shallow Vee. It is claimed that this design permits more tooth area to be in contact under misalignment than any other shape of tooth.

ALL-METAL COUPLINGS

Fig. 83. Double-engagement gear coupling with tubular sleeve. *(Courtesy of The Moss Gear Co. Ltd.)*

A British firm specializing in the design and manufacture of gear couplings with crowned and barrelled teeth is the Fairfield Shipbuilding and Engineering Co. Ltd. These couplings, which can be of flanged or sleeve types, according to requirements, are mainly developed for marine turbines, and, as such, cater for considerable end-float (up to $\frac{1}{2}$ in.), to compensate for thermal expansion of the rotor. In addition to the use of specially treated teeth to combat wear through fretting, these couplings are designed to permit easy access for inspection of the teeth and removal of the pinion or the turbine rotor. Whilst they are built to specifications, typical power ratings range from 1000 to 15,000 hp at 5000 rpm, maximum speeds being in the region of 7000 rpm.

An example of a double-engagement gear coupling with crowned teeth and using a tubular sleeve is shown in Fig. 83. The maximum power ratings of these

Fig. 82. Exploded view of flanged-type double-engagement gear coupling. *(Courtesy of The David Brown Corp. (Sales) Ltd.)*

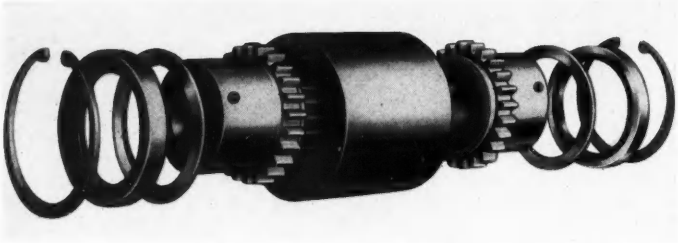


Fig. 84. Exploded view of double-engagement gear coupling with a floating sleeve. (*Courtesy of Crofts (Engineers) Ltd.*)

particular couplings, which are made in England by The Moss Gear Co. Ltd., and which are also available in flanged versions, range from about 3 to 6000 hp per 100 rpm.

Various types of single-engagement and double-engagement gear couplings, with crowned or parallel teeth, according to applications, are produced in the U.K. by Crofts (Engineers) Ltd. Of these designs, a double-engagement type with a floating sleeve is shown in Fig. 84, the sleeve, which is supported on the periphery of the gears, being entirely free to align itself to compensate for angular or axial misalignments. The maximum power ratings of this coupling are from about 900 hp at 31,000 rpm for a $2\frac{5}{8}$-in. coupling to about 41,500 hp at 5200 rpm for a $14\frac{3}{4}$-in. coupling. Single-engagement couplings of this type are also available, with maximum power ratings from about 800 hp at 27,000 rpm to 40,000 hp at 5000 rpm. In addition, special double-engagement couplings of this type are produced by Crofts, e.g., the 54-in. coupling shown in Fig. 85, which is capable of transmitting 90,000 hp at 300 rpm.

For vertical shafts, Crofts supply a special gear coupling, in which the upper shaft is supported by means of a centre diaphragm with a hardened-steel centre disc, lubrication being provided for both the lower and upper halves of the coupling (Fig. 86). Disconnecting-type gear couplings are also produced by Crofts, e.g., that shown in Fig. 87, in which the floating sleeve comprises an outer member and an intermediate member. The outer member on the driving side serves only as a housing for the intermediate member, which transmits torque via shear-pins to the coupling portion on the driven side. Under excessive torque, shearing of the pins permits the intermediate member to free-wheel inside the housing, disconnecting the two halves of the coupling.

As has already been stated, gear couplings are manufactured all over the world, and it is therefore impossible to include in this survey a detailed list of all manufacturers. Thus, for example, to name only a few, in addition to those already mentioned, gear couplings are produced in the U.S.A. by the Koppers Co. (the "Fast's" coupling), the Link-Belt Co., the Dykman Manufacturing Corporation, and the American Flexible Coupling Co., in Canada by the Dominion Engineering Co. Ltd., and in Germany by Malmedie and Maschinenfabrik Buckau R. Wolf. The Wolf coupling, incidentally, is provided with centering rings situated exactly over the centre of the teeth, so that the outer casing can adjust itself without effort to a neutral position. In general, most gear couplings are similar in design, differing only in minor details. A possible exception, however, is the coupling illustrated in Fig. 88. This design, manufactured in Germany by Malmedie, may be considered a gear coupling, insofar as it employs arc-shaped teeth on the inner and outer members; however, torque is transmitted by barrel-shaped rollers located between the teeth. With this arrangement, the coupling can accommodate parallel, angular, and axial misalignments without creating loads on the adjacent bearings. These couplings are available in versions for vertical shafts, and with brake drums, as well as with various overload-protection arrangements. Torque ratings are up to about 900,000 lb-in., with maximum permissible parallel misalignments

Fig. 85. 54-in. double-engagement gear coupling. (*Courtesy of Crofts (Engineers) Ltd.*

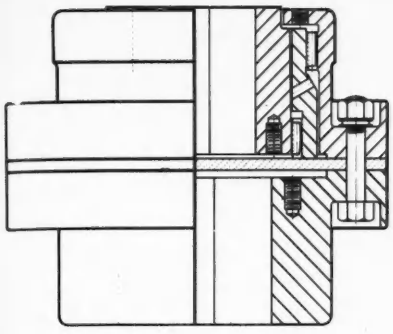


Fig. 86. **Special gear coupling for vertical shafts.**
(*Courtesy of Crofts (Engineers) Ltd.*)

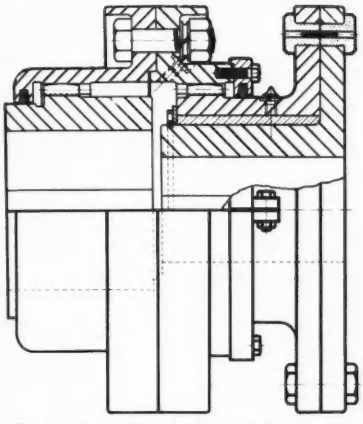


Fig. 87. **Disconnecting-type gear coupling with shear-pins.**
(*Courtesy of Crofts (Engineers) Ltd.*)

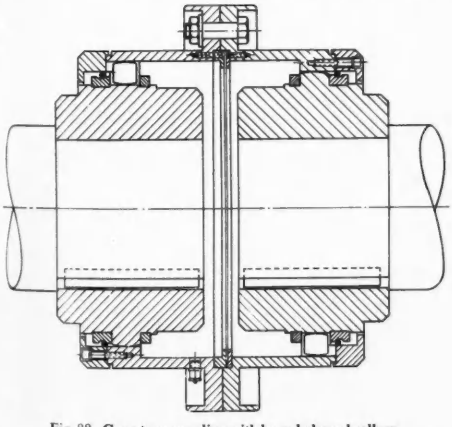


Fig. 88. **Gear-type coupling with barrel-shaped rollers.**
(*Courtesy of Malmedie & Co.*)

from about $\frac{1}{16}$ to $\frac{5}{16}$ in. and maximum angular misalignments up to 1·5 deg.

In general, conventional gear couplings do not have torsional flexibility ; as a result, they cannot be used as damping devices when torsional oscillations are imposed. It is sometimes stated that gear couplings show practically no wear, as they do not incorporate parts which are subjected to repeated bending, tension, and compression. This generalization applies, of course, to the coupling as a whole ; in actual fact, the teeth are loaded in compression and bending, as well as in shear, although they should show very little sign of wear, if adequate lubrication (a necessity in the case of gear couplings) is provided.

(e) *Chain Couplings*

Chain couplings consist essentially of two hubs, located fairly close together and provided with sprockets over which a roller chain is mounted. The teeth are accurately spaced, to ensure that all the rollers in the chain will make uniform contact, thus giving an even load distribution. Clearances are provided between the side-links of the chain and the ends of the sprocket teeth, to permit the hubs to accommodate themselves to any slight angular displacements. The chain is fitted with a master link which is easily detachable, facilitating maintenance and, if required, allowing the shafts to be rotated independently of one another. Installation is particularly simple :—The sprocket hubs are fixed to the respective shaft ends and aligned as accurately as possible, after which they are encircled by the roller chain, whose ends are then linked together by suitable means.

Fig. 89. **Typical example of a chain coupling.**
(*Courtesy of Renold Chains Limited.*)

Although chain couplings can accommodate some end-float and limited parallel misalignment, they are particularly effective in compensating for reasonable angular misalignments. Like other types of couplings, the coupling halves are suitably balanced for use at comparatively high speeds. Designs may vary with regard to the method of attaching the chain ends, but fundamentally all chain couplings are similar and differ mainly in small design details. They are produced by several manufacturers throughout the world, e.g., by Renold Chains Limited and The Moss Gear Co. Ltd. in the U.K., and by the Link Belt Co., the Browning Manufacturing Co., the Chain Belt Company, and the Morse Chain Co. in the U.S.A.

A chain coupling made by Renold Chains Limited is illustrated in Fig. 89. In this design, two chain pinions are

encircled by an easily detachable roller chain, the whole unit being enclosed in a symmetrical cover of bakelite or light metal, depending on size, thereby ensuring complete protection and lubrication (via a grease-gun nipple). The maximum power ratings of this coupling range from about 17 hp at 5700 rpm for a 2¾-in. coupling up to about 1500 hp at 650 rpm for a 24-in. coupling.

Conclusion

In this survey, an attempt has been made to classify flexible couplings into a number of conveniently distinguishable types, to describe their major characteristics, and to indicate their scope and application. During the compilation of this survey, it soon became evident that, so great is the number of manufacturers of these couplings, many of whom produce more than one type, it would be quite impossible to list and describe the products of all makers. While, therefore, every effort has been made to cover the subject as exhaustively as possible, there must inevitably be some omissions, and there can be no doubt that a number of excellent makes in this field have not been covered in these pages. This, of course, must not be taken to mean that they were unworthy of mention, but rather that lack of space has precluded their inclusion. In the same way, when describing couplings by different manufacturers of basically similar characteristics, the more detailed descriptions have usually been confined to couplings by one maker, with a (quite understandable) bias towards the products of British manufacturers.

Finally, it must be stressed that many flexible-coupling applications involve their own particular problems, so that prior consultation and the fullest co-operation between prospective purchasers and manufacturers is nearly always desirable, if optimum results are to be achieved.

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Limits of Combustion Intensity in Heat Engines

By J. J. GROEZE. (From *De Ingenieur, Holland*, Vol. 69, No. 24, June 14, 1957, pp. O.49–O.55, 11 illustrations.)

To a very considerable extent, combustion is to-day a major source of energy release. Combustion is used either to have a heat source which will yield or transfer heat for various applications or which will provide increased pressure or volume of combustion gases for direct mechanical work, in which case heat transfer is a loss of energy that must be avoided. In both cases, i.e., for boilers and furnaces or for heat engines, hydrocarbons, finely distributed and intimately miscible with air, are the main constituents of the fuels to be considered in this article.

In all such applications, the question of the maximum achievable combustion intensity (or energy conversion in kilocalories per unit time and unit volume) arises in connection with the dimensioning of the combustion chamber. When the requirement is to generate heat as rapidly as possible, the chamber is frequently directly surrounded by heat-receiving walls, and the problem becomes that of determining how much heat can be withdrawn during combustion without adverse effects on the combustion process, and how much surface area is required for this purpose. In most cases, the surface area required is so considerable that, when the combustion energy is related to the space available, it is found that the numerical values for combustion intensity are very low.

It is of interest to note that the highest values of combustion intensity occur during the combustion period of high-speed petrol engines, e.g., in racing-car engines with a high specific power output. For instance, if an engine of this class produces 100 hp per litre of swept volume and has a compression ratio of 10 : 1, combustion corresponding to a 100-hp output will occur in a volume of 0·1 litre (i.e., 10^{-4} m^3). If the combustion period is about 20 deg. crank angle of the 720 deg. of the four-stroke engine cycle (i.e., $^1/_{36}$th of the cycle time), and during this period some 2000 kcal per hp per hour are developed, the combustion intensity obtained is $2000 \times 100 \times 36 \times 10^4 = 0{\cdot}72 \times 10^{11}$ kcal/m^{3}hr.

This value, obtained in intermittent operation, is indeed enormous in comparison with those usually found for boilers and furnaces (10^5 to 10^6) or even for modern gas turbines (10^7 to 10^9), and it may well be asked why it is not approached, for instance, by jet engines, particularly since the limitation for petrol and diesel engines is due only to mechanical problems and not to considerations of combustion technology.

The objection has been raised that the combustion-intensity evaluation of the petrol engine should be related to the entire cycle and not only to the 20 deg. of duration of combustion ; this objection is, however, unfounded, since it is easy to conceive that a combustion process of this type can be made to occur continuously and since, in any case, there is no specific cause-and-effect relationship between the duration of combustion and that of the remainder of the cycle. From the point of view of heat transfer to the walls, it is, of course, important that combustion takes place only during 3% of the entire cycle. This is one of the strong points of the piston engine with its cyclic process.

Types of Combustion

Technical combustion processes can be regarded as being composed of three elementary or basic processes. In the first, premixed quantities of fuel and air ignite through spontaneous reaction, and, in the volume of gaseous mixture considered, the explosion is initiated by preliminary reactions occurring throughout the mixture. The characteristic feature of this process is that the explosion takes place suddenly after the mixture has remained for a certain period (the induction period) under favourable conditions.

Fig. 1 shows the successive stages τ of a spontaneous explosion developing during the flow of a mixture through a tube. In the arrangement shown, combustion always occurs at a determined position *A*, of which the distance from the tube inlet depends on the mixture velocity *V*. This process has been particularly studied by Lloyd.[1]

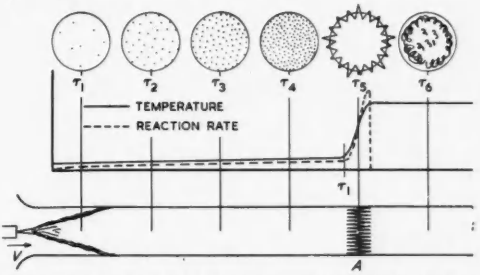

Fig. 1. Successive stages of a spontaneous explosion during flow through a tube.

To simplify description of the reaction mechanism of combustion, the complex preliminary chemical reactions occurring in the temperature range of, say, 250 to 400°C will not be considered here. The entire process can be regarded as a chain reaction, taking place with the aid of radicals which are arbitrarily large fractions of fuel molecules, oxygen atoms, and unstable oxygen compounds, all with free valencies. Also, it can be considered that all these reactions are exothermal, i.e., that they generate heat, during all stages of the process. Viewed in nearly monochromatic light, the spontaneous reaction mentioned above can be seen to be a manifestation in which the reaction rate at a certain stage is entirely determined by the temperature attained at that moment and by one or more concentrations of radicals. Since the reaction causes an increase in temperature, and this temperature rise is associated with an increased concentration of radicals, the entire process accelerates in accordance with

an exponential law. Fig. 1 also shows a typical curve of the reaction rate as a function of time, such as has been calculated, for instance, by Todes[2] from simple mathematical considerations. The initial, nearly flat portion of the curve gives a physical representation of the induction period.

With regard to the stationary combustion in the tube (Fig. 1), it is evident that it could equally be maintained, in principle, if each of the elements of the mixture were packed in a minute cellophane bag ; each element would then still explode by virtue of its own reactions without any influence from neighbouring elements. Given a tube of sufficient length, and assuming that it is made of insulating material, a combustion point will be reached somewhere for each arbitrary value of velocity.

To assess the intensity of this type of combustion, the total volume of tube required for the reaction must be considered. It is found that the shortest induction times for initial temperatures possible in practice (around 600°C) are of the order of 1 millisec. With a tube of 0·01 m^2 cross-section and a velocity of 100 m/sec, a volume of 0·001 m^3 is obtained up to the combustion front. At the temperature mentioned and at atmospheric pressure, the heat of combustion per cubic metre of mixture is about 250 kilocalories, so that the intensity becomes 250,000 kcal/m^3sec, or approximately 10^9 kcal/m^3hr. It is evident that velocity does not enter into the calculated result. The inclusion of the mixing zone, if the latter is properly designed, does not change much in this calculated value.

The process thus described can reach considerable intensities, but it has certain disadvantages. It must be initiated with a reaction velocity which cannot be reached at low initial temperatures. The induction period would then be many times greater, and the space allowed for the reaction (i.e., the length of the tube) would have to be increased in proportion ; moreover, it would then not be certain where the main quantity of heat was generated, and, since it is produced in a very short interval of time, this might involve the further problem of local thermal loading of the walls. In petrol engines, detonation is a manifestation of this type of combustion, and it is known that pistons, etc. are rapidly attacked when it occurs, owing to the chemical aggressiveness of the exploding mixture, which is probably due to a very great local concentration of radicals.

Furthermore, an engine with a periodic combustion process in accordance with this principle would have the disadvantage that its combustion period cannot be fully fixed. Small model aero-engines and the Lohmann engine produced some years ago operate with auto-ignition and have a variable compression ratio.

The other two elementary processes of combustion are based on the fact that, even when the reaction rate is negligibly small at the initial temperature, combustion can be maintained by having the fuel mixture in continuous interaction with the burning mixture. Such reactions must be initiated by auxiliary means, e.g., a spark.

The two types of operating conditions to be distinguished are those occurring with (1) homogeneous mixing of the reaction components ahead of the combustion zone, and (2) separate or direct introduction of these components into the combustion zone. In the first case, the fuel/air ratio is everywhere the same ; in the latter, the complete range of ratios (from 100% fuel to 100% air), depending on time and position, must be considered. As a result, the development of the chemical reactions follows typically different courses in the two cases.

In the first case, starting from a homogeneous mixture of vaporized hydrocarbons and air, only gaseous reaction products are obtained at all stages, with technically usable mixture ratios. If a C-C radical happens to be formed, its life will generally be too short to enable it to sublimate into solid carbon. As a component of the gas it will impart an intense blue colour to the flame.

In the second case, there is every possibility for sublimation into solid carbon particles of the C-C radicals to occur in carbon-rich areas ; consequently, the flame will become lighter in colour, and soot deposition is likely.

As regards maximum possible combustion intensity, there is little to choose between the two cases. Particularly with reactions of negligible velocity, it is frequently possible to arrange for mixing to take place outside the combustion chamber, (e.g., in a carburettor). The possibilities of increasing combustion intensity are so similar in the two cases that the systems can be regarded as fully comparable.

The premixed system, in its simplest form, can be observed in a Bunsen flame, although its most usual occurrence is in the petrol engine, where, as previously stated, the highest intensities are reached. It is therefore of interest to consider both these applications, in view of the simple features of the former and the intensities achievable with the latter.

In considering a flame front, such as occurs in laminar flow, although the oncoming mixture may not show the slightest tendency to react, combustion is maintained through the mechanism of diffusion, with particles from the reaction zone passing into the fresh mixture, and *vice versa*. This diffusion enables heat transfer to take place, increasing the reaction rate of the fresh mixture. In addition, the particles diffusing out of the combustion zone have a high concentration of active radicals, which promote the combustion process. For Bunsen flames burning in free air, the depth of the reaction zone has been found to be of the order of 0·1 mm. The velocity V perpendicular to the flame front, which is termed the normal burning velocity, depends on the type of fuel, mixture proportions, and initial temperature. For a gas flame, e.g., butane in atmospheric air, V is about 0·3 m/sec, so that, taking account of the temperature rise which increases the mean gas velocity, the time taken to pass through the depth of the reaction zone is of the order of 10^{-4} sec.

In view of the very small mean free-path values of the molecules or radicals, it is evident that very many collisions occur during this period in the reaction zone, with heat transfer, diffusion, and exchanges or transfers of momentum. The latter are of interest because radicals, owing to their high chemical activity, always cause similar radicals to be formed in their collisions.

From the zone with a very high temperature towards the end of the reaction, where it is likely that the concentration is highest, and near which the C-C radiation also takes place, no particles diffuse in the short time remaining. Instead, they produce a chain reaction, which can be regarded as a wave of increased radical concentration propagating towards the oncoming flow of gas. The zone over which the influence of temperature extends thus becomes ample and is certainly much greater than it would normally be for the corresponding temperatures, owing to this influx of radicals, which, at a certain critical concentration, can maintain the reaction even without further contact with the hot gases. The zone in which this critical concentration is reached can be regarded as

being the most characteristic section of the flame front.

Thus, the mechanical effect of radicals in flame propagation is similar to that of auto-ignition but differs in two essential respects, i.e., (1) the formation of concentrations of radicals occurs towards the end of combustion and hence at high temperature, and (2) it depends on the composition of the combustion gases. Many light and therefore rapid radicals (e.g., atomic hydrogen) are formed.

In spontaneous ignition, the radicals occur at the beginning of the transformation process, i.e., at comparatively low temperatures, and the chemical composition and structure of the fuel itself have a predominant influence on the quantity and type of radicals formed (e.g., in heptane-benzene). Experience, however, shows that even these very appreciable differences in fuel grade have little influence on the normal burning velocity V, which is much more affected by mixture proportions and the temperature attained. The predominant influence on burning velocity is that of light particles, particularly hydrogen atoms, whose velocity is, on an average, four times greater than that of all other radicals. Acetylene, which releases very considerable quantities of atomic hydrogen in combustion, has a high normal burning velocity. Other hydrocarbons tend rather to split up into larger radicals.

Normal burning velocity also increases as a function of the initial temperature of the fresh mixture, since a rise in this initial temperature will also increase the temperature at the end of combustion. However, comparatively little useful information has hitherto been published in this connection. From tests with kerosene burners at very high initial temperatures, it has been found that by this means normal burning velocity can be increased to about 5 m/sec.

Combustion in Engines

Combustion velocities of even 5 m/sec are completely inadequate to keep motor-car or aircraft engines in operation. Assessments indicate that, even at a very conservative engine speed of 3600 rpm, the flame of a flame front has to travel at a speed of about 75 m/sec. This is achieved by the turbulence in the combustion chamber. The required swirling action can be obtained during suction mainly in the intake passages and along the inlet valve ; in this way, molecular diffusion is supplemented by turbulent diffusion, which is far more effective. Turbulence causes a breaking up of the flame front and an increase in its surface area ; it also probably enables critical radical concentrations, even without direct contact with the flame front, to carry the reaction further. The remarkable thing is that the inlet velocity is of the order previously mentioned ; as it varies with the running speed of the engine, this may explain to some extent why burning velocity increases with running speed. Moreover, it may be noted that the inlet eddies are somewhat damped out during the time elapsing between admission and the beginning of combustion. This time interval becomes shorter when running speed is increased, so that it can be said that the state of turbulence during combustion definitely increases at least proportionally with running speed.

In endeavouring to attain maximum outputs at very high speeds (10,000 rpm, or even higher for racing-car engines), there is no need for further arrangements to ensure sufficiently rapid combustion. A modified design is necessary only when particularly high flexibility of the engine is desired, this being obtained by increasing the

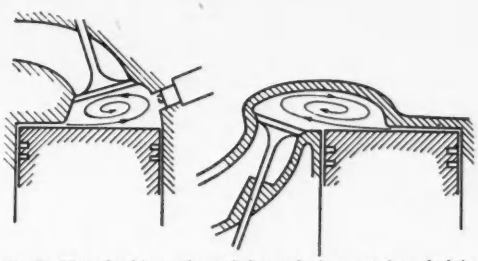

Fig. 2. Methods of increasing turbulence of mixture at the end of the compression stroke.

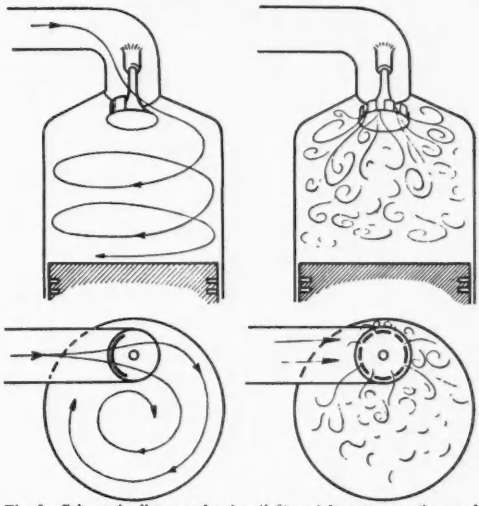

Fig. 3. Schematic diagram showing (*left*) mainly rotary motion, and (*right*) random motion of inflowing gas.

turbulence at low running speeds, e.g., by increasing the swirl of the charge at the end of the compression stroke (Fig. 2). It has been found in tests by van Hoogstraten[3] that, with the same energy in the inlet eddy, a motion which is principally rotary (Fig. 3) is more effective than a random motion, probably because less of this energy is lost through friction.

Curves of the duration of combustion (in degrees of crank angle) plotted against mixture strength are practically flat over the entire range of fuel/air ratios tested, indicating that combustion time is to a considerable extent independent of mixture proportions. This is as expected, since the diffusion of flame through the combustion chamber is sustained by turbulence, i.e., by a mechanism which is independent of the type of fuel and mixture proportions. However, it should be noted that combustion time is affected by the type of inlet valve used. It has been found that an approximately 45-deg. crank angle is required with a normal valve, 40 deg. with a masked valve with eight peripheral slots, and from 30 to 38 deg. with a rotating-shield type of valve.

The accelerating effect of turbulence, however, causes only a "global" or large-scale transport of the flame elements. In all the stagnant regions between and inside the eddies, molecular diffusion still has to complete the process. As the smallest eddies are of the order of 1 mm, the time lost for this final diffusion is only 0·2 millisec, so that this period, which is also variable, is again small in comparison with the total duration of flame propagation.

The combustion times of petrol engines have been mentioned above as being of the order of milliseconds, whereas the total reaction time of the Bunsen flame front is in the region of one-tenth of these values. However, in view of the greater densities involved, there is every reason to suppose that, in petrol engines, the development of the reaction itself is at least equal to, or even more rapid than, that in the Bunsen flame, so that the duration of the reaction at any point is considerably smaller than that of flame propagation.

Limiting Values

The combustion intensity of the petrol engine can be represented as a single point in a wide range of possible values. Some time ago, Longwell[4] tried to find a general limit for intensities, his method involving the use of a bomb with six inlet tubes through which a mixture of propane and air could be injected, and the reaction products withdrawn through six other passages. In each of the tests carried out, he endeavoured to determine the limiting values of injection velocity for which the reaction, on the basis of exhaust-gas analyses, was still complete, and found the corresponding values for reaction intensity, the variable parameters being pressure and mixture ratio.

Longwell regarded the combustion in his bomb as a steady-state reaction. It is also possible to consider it as a case of progressive combustion, i.e., with propagation occurring towards the oncoming strongly turbulent jet streams, and it is of interest to see whether the same considerations as for the petrol engine can be applied in this instance. In his experiments, Longwell found reactions in which velocity increased as the square of pressure. Since the potential energy content already varies proportionally to pressure, the reaction rate must also have varied with pressure, and this was probably related in some way to the completion time of the reaction. Considering the total time, it can be seen that it consists of turbulent diffusion time, local molecular diffusion time, and local reaction time. The first of these is independent of pressure, and the second decreases with increasing pressure, though only slightly at atmospheric and the highest pressures.

Longwell reached limiting values corresponding to very high transformation intensities at high pressures, i.e., $3{\cdot}5 \times 10^{11}$ kcal/m^{3}hr at 10 atm. At 1 atm his value was therefore about $3{\cdot}5 \times 10^9$ kcal/m^{3}hr, which is about 20% of the observed intensity in the reaction zones of the Bunsen flame. It appears, therefore, as if, in Longwell's experiments, transport times are reduced to the order of magnitude of the reaction time, so that very little more can be achieved. This can be explained by the fact that in his tests the scale of turbulence, i.e., the size of the largest eddies, was sufficiently large, in comparison with other distances to be covered, whereas the intensity of his turbulence was such that only very small distances had to be covered by molecular diffusion in small stagnant regions.

A schematic comparison of the three cases, i.e., the laminar Bunsen flame, the petrol engine, and Longwell's bomb, can be made, and it is interesting to note that in each case different phases of the process determine the transformation velocity, these being molecular diffusion, followed by turbulent diffusion and local reaction, and finally, at a density comparable with that in an engine, all three phases to approximately the same extent.

Both in Longwell's bomb and in the petrol engine the extent of turbulence is very great ; if it is considered in relation to the inflow velocity of the charge, the degree of turbulence reaches a calculated value which is infinitely large, since the motion is brought to a standstill while the turbulence remains. In the combustion chambers of gas turbines and ram jets this never occurs in practice, because the generation of such strong turbulence requires an amount of energy which cannot be provided by the considerably lower energy level of these engines. These processes operate with much lower degrees of turbulence, e.g., 5 to 10%, but the transformation intensities obtained are then also correspondingly lower.

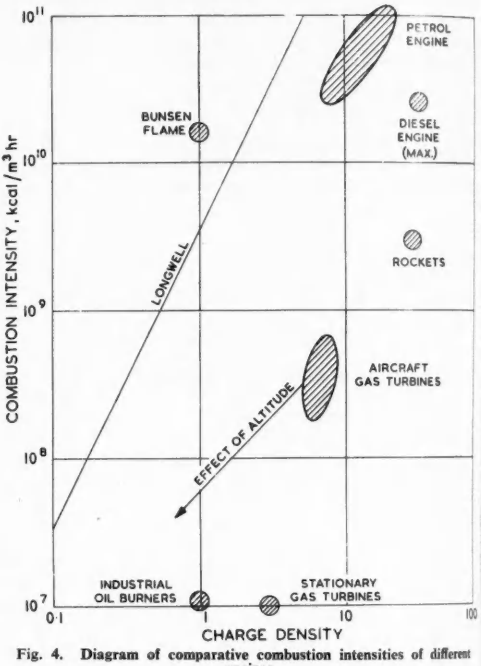

Fig. 4. Diagram of comparative combustion intensities of different engines.

Fig 4 gives a comparison of values of combustion intensity reached in various engines, plotted against charge densities. The solid line represents the maximum values indicated by Longwell for a theoretically correct mixture of propane and air. Above it can be seen the value found for the reaction zone of a Bunsen flame. The petrol engine has the highest value, and the best value obtained for a diesel engine (the Napier "Nomad" with a specific output of approximately 100 hp per litre of cylinder capacity) is situated somewhat lower. This is not primarily because mixture in diesel engines takes place during combustion but rather because, owing to considerable turbulence and high densities, the limit of permissible thermal loading is reached earlier.

The intensities in aircraft gas turbines are situated much lower in the diagram, owing mainly to the smaller degree of turbulence employed and also very probably because, with continuous combustion, the limit for permissible thermal loading of the walls is reached more rapidly. Density decreases with increasing altitude, with a corresponding decrease in combustion intensity at constant maximum temperature. The arrow indicates that there is a tendency to approach the straight line of limiting values found by Longwell, and it is a fact that at high altitudes a tendency towards incomplete combustion

has been observed in various combustion chambers.

Rockets are located higher up in the diagram, because in most cases, owing to the pressure to which the reactants are subjected, the intensity of turbulence in the combustion chamber is very high, and the limit is determined primarily by considerations of thermal loading. The loadings of typical stationary combustion turbines are very conservative, and are equalled by some modern industrial oil burners, in which intensive turbulence is employed. Data on ram-jets are not known to the author.

Thus, according to the present state of knowledge, it can be considered that the extreme limits are determined by the scale and intensity of turbulence employed, and by density. The scale of turbulence, together with the supply velocity of the mixture, determines the distance which can be covered per unit of time and hence the magnitude of the space which can be penetrated per unit mass of mixture. The intensity of turbulence determines the dimensions of the minimum size of eddies, and therefore also the time which is required for the essentially slower process of molecular diffusion. Density determines the velocity of the development of the reaction at each point.

For piston engines, the variable factors available are amply sufficient for further development. For gas turbines and ram-jets the problem is to find means of producing turbulence with a minimum expenditure of energy. In this connection, the problem of wall life becomes increasingly difficult.

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GERMANY

Unbalance and Centrifugal Forces in Grinding Wheels

By O. Kienzle and H. Münnich. (From *Werkstattstechnik und Maschinenbau*, Vol. 47, No. 2, February 1957, pp. 69–73, 7 illustrations.)

Unbalanced grinding wheels excite vibrations due to centrifugal forces. For extremely fine surface finish, therefore, unbalances and their causes and effects must be considered more closely, and tolerances for economic working and manufacture must be established.

Varying density or asymmetrical dimensions may displace the centre of gravity of a grinding wheel of weight G by an amount r_s from the axis of rotation (Fig. 1) and thus produce a centrifugal force

$$P = r_s \omega^2 G/g$$

The product Gr_s is the static unbalance, which may also be characterized by an "unbalance weight" $A = 2Gr_s/D_a$ assumed to be applied to the outside diameter D_a of the wheel in such a manner that it is able to balance the disc. Another definition is the "relative unbalance weight" $\varphi = A/G$. A wide grinding wheel may have in addition a dynamic unbalance if its axis of inertia connecting the centres of gravity in various planes differs from the axis of rotation. However, this dynamic unbalance need not be considered, unless the width of the disc is greater than, say, one sixth of its diameter[1], and will be ignored here. Static unbalance can be eliminated by a balancing weight A' applied at a radius r_A on the side opposite the original centre of gravity S_o (Fig. 2), so that

$$A' = Gr_s/r_A$$

If possible, the weight A' should be applied equally on either side of the disc, so that it cannot create a new dynamic unbalance.

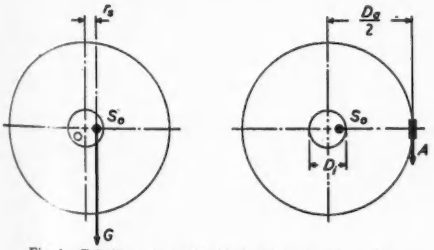

Fig. 1. **Grinding wheel with displaced centre of gravity.**

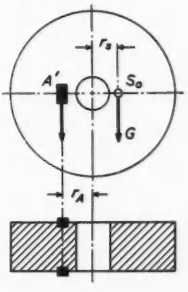

Fig. 2. Elimination of static unbalance by means of a balancing weight.

The unbalance of the wheel undergoes changes during manufacture, delivery, fitting, and actual operation. Structural differences or geometrical irregularities during manufacture cause a manufacturing unbalance which must be limited, because grinding quality must not suffer and the internal stresses due to the unbalance itself or to the requirements for balancing must not impair the safety of the wheel. The manufacturer usually works to self-imposed tolerances and passes only those wheels which satisfy them ; these are then balanced by additional weights, e.g., lead-weighted bushes or grinding matter added by impregnation or adhesion.[2] The remaining unbalance is the unbalance as supplied, and, for manually operated grinding rigs without balancing possibilities, it must be small enough not to disturb working. If the wheel is to be held in mounts with balancing weights in tee-grooves or threaded holes for fine balancing, the unbalance as supplied must be limited to within the balancing range of these mounts. Static or dynamic balancing of the mounted wheel then removes not only

the unbalance of the wheel as supplied but also any new unbalance due to eccentricity of mandril and wheel or any change in unbalance caused by a first dressing operation. The unbalance now remaining when the wheel is fitted into the machine may be called the mounting unbalance, and must be limited in accordance with the expected quality of the ground surface. It can hardly be affected by any additional eccentricity due to fitting, because the receiving end of the spindle is tapered ; nor can absorption of humidity change this unbalance, since the coolant is soon ejected centrifugally. However, the mounting unbalance can change with wheel wear if, e.g., one side of the circumference loses denser material than the opposite side. Thus, the unbalance at any stage of operations, or the operational unbalance, may have to be eliminated from time to time by a change in the balancing weights so that the wheel continues to run with a minimum of vibration.

The effect of varying density or of irregularity of shape on the change of the operational unbalance during wear, i.e., with decreasing diameter of the grinding wheel, can be visualized (Fig. 3) if, for instance, it is assumed that a sector of 90 deg. differs in density from the rest of the wheel (Fig. 3a), or that the sides are not parallel (Fig. 3b), or that the wheel is mounted eccentrically, owing to too large a bore (Fig. 3c). The graphs in Fig. 3 show the total remaining unbalance Gr_s plotted against relative wheel wear, i.e., the ratio of worn wheel diameter D_w to original wheel diameter D_o. An initial " relative unbalance weight " of $\varphi = 0{\cdot}2\%$ has been assumed in each case. In Fig. 3c the unbalance is reduced to negligible proportions after the first dressing operation ; in Figs. 3a and 3b it diminishes gradually. With constant speed the centrifugal forces are proportional to the unbalance but, if the circumferential speed is kept constant, i.e., if the disc is speeded up as the grinding-wheel diameter decreases, then the centrifugal forces diminish more slowly than the unbalance.

The unbalance would be negligible throughout the life of the grinding wheel if the balancing mass could be attached in such a way that with decreasing diameter it would always be removed to the same extent as the original unbalance. This can be achieved if the balancing mass is distributed over a certain area, e.g., by the addition of grinding matter. However, such a method is not feasible with fine-grain wheels and is difficult and expensive, if accurate balance is required (Fig. 4a). There remains local balancing by means of weights in the wheel mounts. Evidently, if a new wheel is balanced completely by such weights, then, as the original unbalance wears away, the weights themselves will increasingly create a new unbalance.

Suppose that a wheel is balanced only once, when it is taken into operation with an original outside diameter D_o. If the balancing weights in the mounts are so dimensioned and arranged that complete balance is obtained at the start, i.e., if the unbalance of the disc is opposed by a roughly equal unbalance of the added weights, then at the end of its life the wheel, now of final diameter D_f and with its original unbalance worn away, will be thrown out of balance by the effect of the weights. The final unbalance will be nearly equal to the original unbalance without weights, so that, in the end, balancing has been of no avail (Fig. 4b). If throughout its life the wheel runs at a constant rotational speed, the centrifugal force will eventually be equal to the centrifugal force which the original unbalanced wheel would have had at the start. If, however, the circumferential (grinding) speed is kept constant and the wheel speeded up as its diameter decreases, then the centrifugal force (due to the weights) will be eventually nearly $(D_o/D_f)^2$ times larger than the initial force without balancing weights ; in this case, initial balancing has not improved but has worsened the situation.

Hence, a wheel running at a constant rotational speed throughout its life will be most effectively balanced if

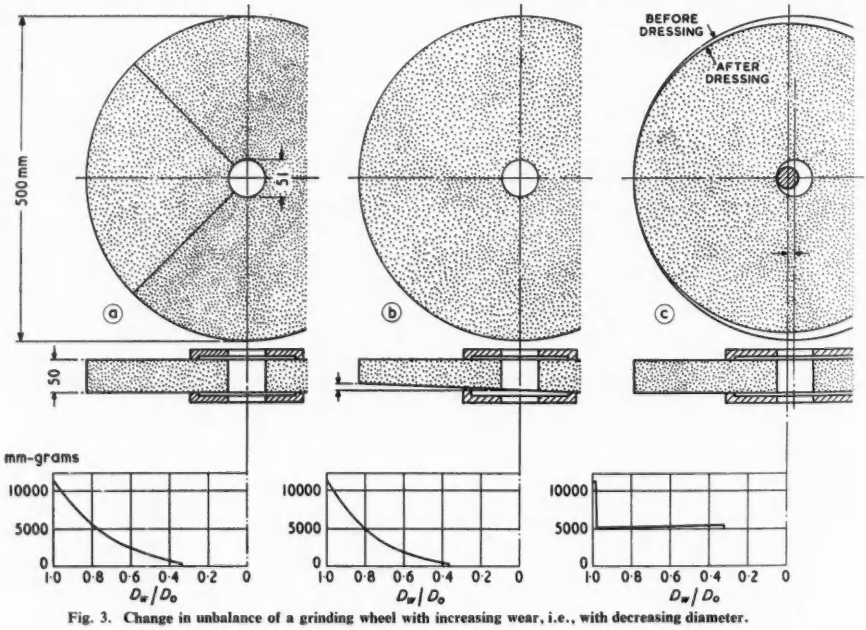

Fig. 3. Change in unbalance of a grinding wheel with increasing wear, i.e., with decreasing diameter.

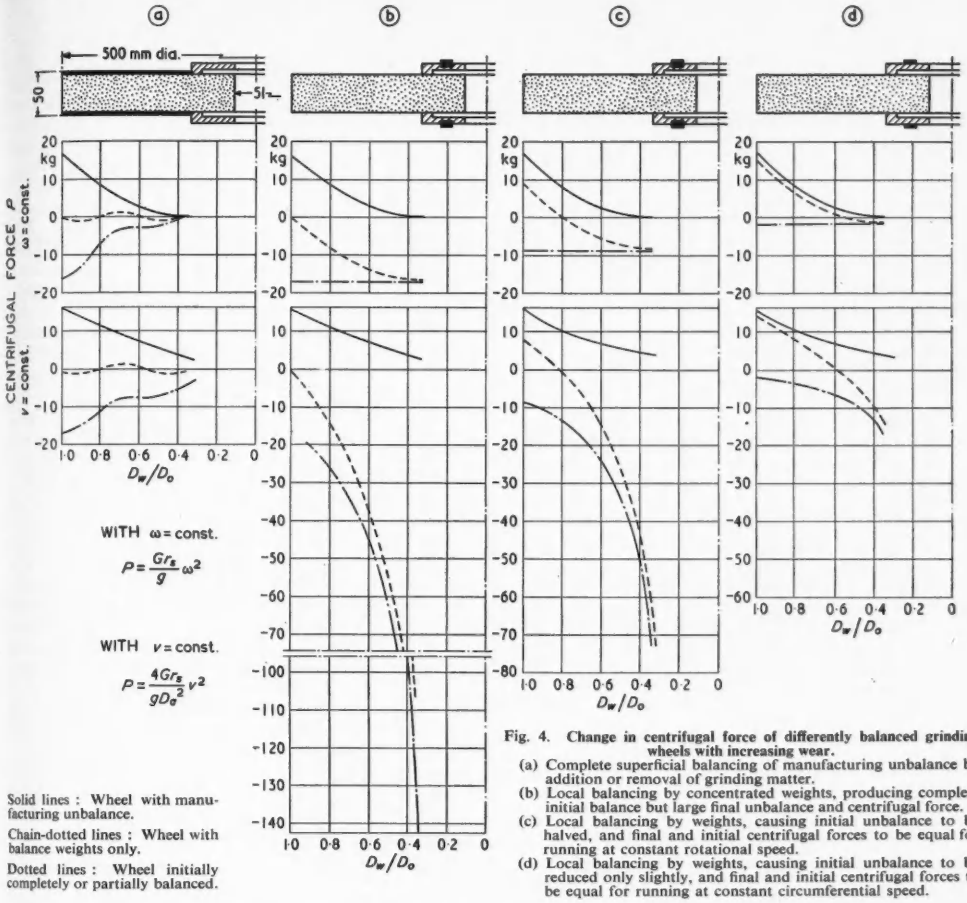

Fig. 4. Change in centrifugal force of differently balanced grinding wheels with increasing wear.

(a) Complete superficial balancing of manufacturing unbalance by addition or removal of grinding matter.
(b) Local balancing by concentrated weights, producing complete initial balance but large final unbalance and centrifugal force.
(c) Local balancing by weights, causing initial unbalance to be halved, and final and initial centrifugal forces to be equal for running at constant rotational speed.
(d) Local balancing by weights, causing initial unbalance to be reduced only slightly, and final and initial centrifugal forces to be equal for running at constant circumferential speed.

initially only half the unbalance is opposed by the added weights (Fig. 4c). The unbalances and centrifugal forces at start and finish will then be equal. A wheel running at constant circumferential speed, on the other hand, must be balanced initially only to a small fraction[3], roughly equal to $(D_f/D_o)^2$, if initial and final centrifugal forces are to be about equal (Fig. 4d). In all cases, a relative unbalance weight of $\varphi = 0{\cdot}2\%$, due in equal parts to variations in density and shape, has been assumed.

For precision grinding it is not considered sufficient if the wheel is balanced only once during its life ; it must be rebalanced when the centrifugal force exceeds a certain permissible value recognized by the onset of running disturbances or of increasing roughness. Repeated balancing is best arranged so that the necessary number of balancing operations is as small as possible, this number depending to some extent on the cause of the initial unbalance. With wheels running at a constant circumferential speed, the number of balancing operations also depends on the size of the balancing weights ; the heavier these are the more rapidly the centrifugal force changes with decreasing diameter, i.e., with increasing angular velocity. Consequently, precision grinding wheels should not be balanced by localized weights during manufacture, if they are to be used at constant circumferential speed.

These considerations supply a basis for a standardization of limits for the unbalance of grinding wheels. Safety requires limitation of the manufacturing unbalance and of balancing arrangements in recesses of the grinding wheel itself. For grinding quality and quiet operation the wheels can be divided into three groups, i.e., (1) wheels which are not balanced ; (2) wheels which are balanced only once before mounting in the machine ; and (3) precision wheels which are rebalanced as soon as the centrifugal force exceeds a permissible limit.

The manufacturing unbalance and unbalance as supplied should be standardized separately for these three groups. The balancing weights for the wheel mounts should be chosen in accordance with the standardized limits for the unbalance as supplied.

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U.S.A.

Recent Developments in Magnetic-Force Welding

By E. J. Funk. (From *The Welding Journal*, Vol. 36, No. 6, June 1957, pp. 576–582, 14 illustrations.)

The magnetic-force welder is fundamentally a conventional direct-acting ram-type spot welder which has undergone design changes to accommodate an electromagnet for the application of electrode pressure. The flow of current is controlled by a conventional electronic timer and sequence control, and passes into the primary of the welding transformer, which is of a special design, having components sized in such a manner as to meet the requirements of magnetic-force welding. The secondary current is directed to a split-iron-core magnet and makes one turn around the stator, after which it passes through a flexible laminated roll lead to the upper electrode holder. The return path is completed after the current passes through the weldments and lower electrode.

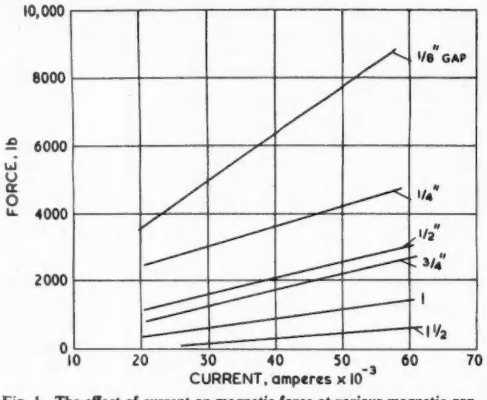


Fig. 1. The effect of current on magnetic force at various magnetic-gap settings.

A small air cylinder is provided to lift the electrode to the desired opening for work insertion and removal, and to provide clamping pressure. The ram is of the quill type, rigidly supported by linear-action ball bushings, which allow extremely fast follow-up with a minimum of mechanical friction. The centre portion of the ram is threaded to permit adjustment of the force collar which supports the magnet armature. The lower end of the ram is securely attached to the upper electrode holder, which is usually of an ejector type. The lower knee is a high-conductivity copper casting which is provided with a tee-slotted platen for universal die mounting and which is supported by the face-plate and an adjustable knee-jack.

The behaviour of the force in a magnetic welder is controlled by the same physical laws governing the behaviour of any electromagnet. These laws state that, in a magnetic circuit, the flux is directly proportional to the impressed magnetomotive force, and inversely proportional to the reluctance of the magnetic circuit. Since in this instance the force magnitude is under consideration, it may be said that the force is a direct result of the flux in the armature and is therefore synonymous with it. The magnetomotive force of the circuit is proportional to the strength of the current in the conductor and to the number of times the current encircles the magnetic path, and is usually expressed in terms of ampere-turns. Since the welding-current conductor has only one turn around the core, the magnetomotive force may be considered to be a direct function of current magnitude. The reluctance of a magnetic circuit is proportional to the length of the circuit and inversely proportional to the cross-sectional area, and depends on the magnetic properties of the materials composing the circuit. It may be concluded from theory that the magnetic force is directly proportional to current and inversely related to the air gap. Magnetic-force welders are designed so that the air gap between the armature and the stator of the magnet may be adjusted in order to allow variation of the reluctance of the magnetic circuit.

A set of experimental curves obtained from a study of the force characteristics exhibited by a 75-kVA welder is shown in Fig. 1, in which maximum force is plotted as a function of welding current for several magnetic gaps. It will be noticed that actual performance is in good agreement with theory, i.e., that magnetic force has a linear relation to current and is reduced by an increase in gap.

Information of this type is useful during set-up procedures, as it gives an indication of how adjustments affect welding conditions. As the rate of heat generation in resistance welding is proportional to the product of the resistance of the welding circuit and the square of the current, it can be seen that adjustment of transformer taps or phase-shift settings, in such a manner as to increase the current magnitude, would have two effects on the welding situation, i.e., (1) the rate of heat generation would increase as the square of the increase in current magnitude, and (2) the electrode force would increase, reducing the electrical contact resistance and at the same time improving the conditions for heat transfer away from the work. Both of these effects would cause a reduction in the apparent heat present at the weld zone. Experience has shown that the decrease in heat, due to the increase in force which accompanies a current increase, is small with respect to the increase in heating caused by the additional current and, for the most part, need not be considered during set-up procedures.

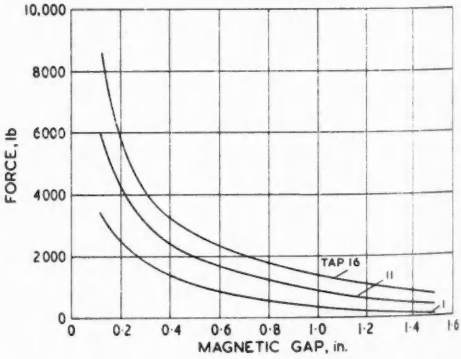


Fig. 2. The effect of magnetic gap on force magnitude for several transformer taps.

The effect of magnetic gap may be observed in Fig. 2, which shows maximum force plotted as a function of gap for several transformer taps. The curves were plotted from the same data used for the set of curves in Fig. 1, and are intended to show the manner in which force varies when all machine settings remain constant, the only variable being magnetic gap. It is interesting to

note that the experimental results indicate that the forces tend to respond in accordance with Coulomb's law for point poles, i.e., that the force of attraction is inversely proportional to the square of the distance between poles.

This information is useful from the standpoint of operating the equipment, because it shows that small adjustments of magnetic gap, in the range of $\frac{1}{8}$ in., will produce large differences in the force, whereas much larger adjustments, in the range of 1-in., will result in practically no change in force.

Magnetic-gap opening has one effect which is not readily observed in the curves shown in Figs. 1 and 2, i.e., the slight increase in secondary current which accompanies an increase in gap. It has been observed, for example, that current increased from 56,000 to 60,000 A during the course of investigation at the top transformer tap, as the gap was changed from $\frac{1}{8}$ to $1\frac{1}{8}$ in. The relative magnitude of the increase is of no practical importance insofar as operation is concerned.

The Delayed-Forge System

The early approaches to the development of magnetic equipment involved the use of rectified three-phase welding current which yielded a force with a wave-form having considerable ripple and a continuous force application throughout the heating cycle. The development of single-phase equipment was delayed, because early experience led to the belief that the relaxation of magnetic force, which occurred as the current wave changed polarity, would have an unsatisfactory effect on the weld. However, in the course of experimental welding, it was found that the required heating times could be very short, often only one-half cycle, and the development trend turned to single-phase equipment. At this point, it was found that welding schedules having these extremely short heating periods required some rather high values of current, as compared with conventional spot-welding applications. The original single-phase machines were ultimately called upon to perform welding tasks for which they simply did not have enough capacity. It was out of the need to extend the capacity of small machines and enable them to encompass a greater range of work that an idea for controlling the time of application of magnetic force was evolved. It was thought that, if a period of heating time was allowed to take place before the electrode pressure due to magnetic action was applied, a lower current would be needed to accomplish the same amount of heating and thereby the power requirements of the machine would be reduced.

The principle is simple, involving the use of a delay magnet, energized from a direct-current source, which withholds the application of electrode force until the force due to the welding current becomes of sufficient magnitude to pull the armature away from the holding poles. In the actual design the delay magnet has a breakaway strength of something of the order of 50 lb and serves to hold the armature a small distance above the collar on the welding ram. As the welding current builds up, the holding force is rapidly overcome and the armature is accelerated for a short distance before coming into contact with the ram. The amount of free travel experienced by the armature, from the time it leaves the delay magnet until the time it bottoms on the ram collar, is known as the " delayed-forge gap ". The delayed-forge gap should not be confused with the basic magnetic gap, which is determined by the position of the ram collar, with the work in place, and refers to the distance between the armature and the stator poles. The two quantities are independently variable, although the resulting force wave is a function of both.

A laboratory investigation has been conducted to determine the effect of the delayed-forge system on the resulting electrode pressure wave with respect to time and magnitude. In this connection, Fig. 3 is a graphical tabulation of data concerning the effect of various delayed-forge gap settings on the time and magnitude of the resulting force wave. The peak force is plotted as a percentage of the maximum value of force obtained without delayed forge. Delay time is plotted in units of electrical degrees (360 deg. per cycle at 60 cps) and represents the time difference between the points of force initiation under normal magnetic-welding conditions and under delayed-forge conditions. Examination of the curves reveals that increasing the delayed-forge gap results in a corresponding increase in delay time. However, an increase in gap does not necessarily dictate a corresponding increase in force magnitude. It can be seen that a point of diminishing return is reached, after which an increase in delayed-forge gap yields a decrease in force amplitude.

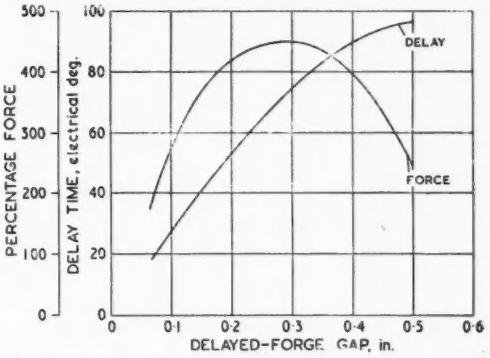


Fig. 3. The effect of delayed-forge gap on relative force magnitude and delay time.

The delayed-forge system permits a long heating period to occur, during which the weldments are subjected only to the clamping force provided by the air cylinder. The heating period is terminated by the impact-force wave which completes the weld cycle. It has also been observed that the magnitude of the force wave varies considerably, depending on the force system employed. The higher values resulting from the use of the delayed-forge system are due to the energy storage which occurs as the magnet armature traverses the delayed-forge gap.

It has been found in practice that the delayed-forge system accomplishes its original purpose of reducing the power requirements for certain applications, and the feature is included in present-day designs. The system offers an additional advantage which is a direct result of the increased force magnitude. In the welding of electrical contacts, the highly objectionable condition of clearance existing between the contact and its arm after welding often arises. The problem of obtaining complete set-down has been satisfactorily solved in a number of production cases through the use of the delayed-forge system.

" Magna-Flash " Welding

Essentially, " magna-flash " welding involves the use of a small projection which is caused to disintegrate and assist in the formation of an electric arc at the

interface of the weldments. Once arcing has occurred, the weldments are brought into intimate contact and allowed to cool. The final joint is welded over its entire area and exhibits very fine electrical and physical characteristics. The narrow nature of the entire heat-affected area, typical of magna-flash welding, indicates that the process involves an intense heating over an extremely short period of time.

A photomicrograph of a magna-flash weld in a typical automotive valve lifter is shown in Fig. 4. In the upper portion can be seen the unaffected cast iron, which consists of graphite flakes and carbides, dispersed in a matrix of fine to semi-coarse lamellar pearlite. Progressing downwards, a heat-affected zone can be observed in the cast iron, consisting of 10% eutectic ledeburite. about 0·003 in. thick. The diffusion zone is a martensite layer which varies in thickness from almost zero to 0·01 in. and is adjoined by a heat-affected zone in the steel body, identifiable by patches of martensite and blocky ferrite. The unaffected structure of the valve lifter is of steel with fine lamellar pearlite existing in a ferrite matrix.

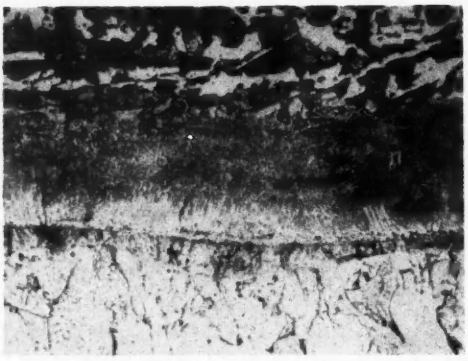


Fig. 4. Photomicrograph of a magna-flash weld in an automotive valve lifter. (× 500)

An understanding of the principles of the magna-flash process requires consideration of a different set of variables than those normally considered in connection with resistance welding. Of course, the primary fundamentals of current, pressure, and time are still just as important in magna-flash welding as they are in other resistance-welding processes, but it may be convenient to think in terms of electric arcs and fast follow-up, rather than of a heating process which depends upon the flow of an electric current through the resistance of the welding circuit.

Examination of oscillograms of the primary current wave associated with a magna-flash weld reveals a peculiar decrease in current, occurring soon after the start of initial current flow. This momentary decrease is soon followed by a new increase, which continues to a maximum and then drops off, as would be expected of a sine wave. Since the wave form results from a half-cycle weld time, the initial current pulse is followed by a surge of current in the opposite direction, induced by the collapsing transformer field, and gradually decays to zero. The observation of the unusual " notch-effect " occurring in the leading portion of the current wave has resulted in further investigation of the current and voltage relationships involved in the magna-flash process. Study of the voltage variation, with respect to current, reveals that there is an instantaneous increase in voltage which becomes stabilized and remains nearly constant throughout the duration of the notch effect observed in the current wave. The voltage experiences a very rapid decrease almost simultaneously with the second increase in current.

The behaviour of voltage and current is explained in the following manner :—At the commencement of welding, the two parts are separated by means of a small projection. When the initial flow of current occurs, this projection is heated so rapidly that it disintegrates and assists in establishing an electric arc between the weldments. As the arc is established, the rate of rise of current is interrupted, and current flow is controlled by the arcing conditions. During this period, the voltage first approaches the open-circuit voltage of the transformer and then stabilizes at a value which depends, to some degree, on the materials being welded. As the parts being welded are brought into contact with each other, the arc is extinguished, and resistance to the flow of current approaches short-circuit conditions. The voltage across the weldments drops off and is accompanied by a sharp rise in current. Theoretically, the heating is complete when the arc is extinguished, as indicated by the decrease in voltage, and all that remains to be done, in order to complete the weld, is to cool the parts under light clamping pressure.

It is interesting to note that the time interval from the initial heating of the projection until the voltage drop occurs, indicating the completion of arcing, is of the order of 0·0032 sec. The mechanical action of the welder must be capable of bringing the weldments into contact within this period of time, in order to effect a successful weld. The ability of the welder components to remain in contact with the work and to apply pressure throughout the welding cycle is usually referred to as " follow-up action ". Follow-up action in a magnetic welder can be controlled, to a great extent, by adjustment of the magnetic gap.

In practice, it has been found that the tooling for magna-flash welding must be capable of firmly gripping the weldments, in order to prevent the surface burning hich would result from a loss in contact pressure between the weldments and the electrodes. The best results have been obtained with tooling designed so as to depend upon the action of the magnetic ram for follow-up and to provide an independent source of clamping force which will ensure that the weldments remain undisturbed until cooling is completed.

Conclusions

A close examination of the behaviour of magnetic-force equipment shows that its principal feature is the ability to produce accurately and reproduce certain combinations of heat, time, and pressure which are not obtainable with conventional resistance-welding equipment. These combinations of variables have been successfully applied in a number of production situations and are credited with producing greater weld consistency and improved electrode life, and with the ability to join materials previously considered unsuitable for resistance welding. As a result, the field of resistance welding has been further extended as a metal-joining method which offers reduction of production costs and product improvement. However, the exact functioning of magnetic welders to bring about the solution of many difficult welding problems is not clearly understood, and many facets of the field of magnetic-force technology remain to be explored and explained.

The Governing of Diesel Engines for Vehicles

By T. Szirtes. (From *Acta Technica Academiae Scientiarum Hungaricae*, Vol. 16, Nos. 3/4, 1957, pp. 233–254, 6 illustrations.)

The stability of a governed engine can be assessed from its torque/speed characteristic, i.e., large changes in torque should correspond to small variations in engine speed on either side of the controlled-speed value. The effectiveness of the governor depends on various design features, and the object of this paper is to consider these features and their effects.

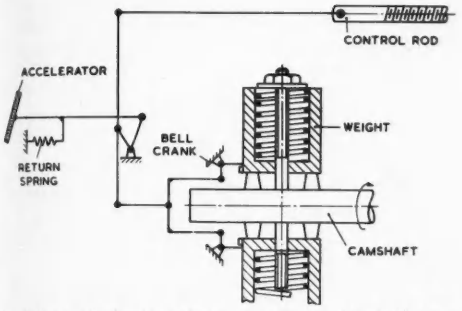


Fig. 1. **Schematic drawing of fuel-pump governor and control linkage.**

An illustration of a centrifugal governor arrangement is given in Fig. 1, in which the outward travel of the weights, due to centrifugal force, is resisted by springs. Let s_0 = distance from centre of gravity of weight to axis of rotation, x = displacement of weight from its initial position, e = spring preload setting, K = spring stiffness, and m = mass of spring (distances in metres, stiffness in kg/m, and mass in kg sec^2/m). At any speed N [rpm] of the fuel-pump camshaft, the centrifugal force is $m\omega^2(s_0 + x)$, and it is opposed by the spring force $K(e + x)$. These can be written as an equation, which can be solved for the displacement x, so that

$$x = (s_0N^2 - ke)/(k - N^2) \quad .. \quad .. \quad .. \quad (1)$$

where $k = (K/m) \times (60/2\pi)^2 = (K/m)/0{\cdot}01096$. Each displacement x also corresponds to a certain reduction in the quantity of fuel injected. A plot of x against camshaft speed N is shown in Fig. 2, in which the curve of eq. (1) is of the parabolic type. The stability limit (when x tends to become infinite) is when $N = \sqrt{k}$, i.e., when the denominator of eq. (1) is zero. At $N = 0$, $x = -e$, but, owing to the precompression of the spring, x cannot be less than zero. Hence, with $x = 0$, eq. (1) gives

$$N_0 = \sqrt{ke/s_0} \quad .. \quad .. \quad .. \quad (2)$$

This is the " control-point speed ". Between N_0 and the origin, the true governor characteristic is not parabolic but follows a straight line along the N-axis. The characteristic thus has a discontinuity at N_0.

Furthermore, each preload setting e_1 or e_2 of the springs corresponds to a different characteristic and a different control point N_{01} or N_{02} (eq. (2) and Fig. 2). The stability limit, however, is independent of preload and is exclusively determined by the stiffness and mass values, which give $\sqrt{k}$. In reality, of course, this theoretical value of $N_\infty = \sqrt{k}$ cannot be attained, since the spring becomes coil-bound at some finite value of displacement x_c.

If eq. (1) is solved for N in terms of x, the result is

$$N = \sqrt{k(x + e)/(x + s_0)} \quad .. \quad .. \quad (3)$$

Now, if the momentary camshaft speed is, say, N_a, it will correspond to a spring deflection $x = a$ (considering curve (1) in Fig. 2). This value a can be substituted in eq. (3) to obtain N_a. The difference in speed is $\triangle N = N_a - N_0$, where N_0 is obtained from eq. (2). By taking the derivative of $\triangle N$ with respect to e, it can be shown that $\mathrm{d}\triangle N/\mathrm{d}e$ is always negative, i.e., $\triangle N$ is reduced when the preload is increased (compare curves (1) and (2) in Fig. 2). Since N_∞ is unchanged, this means that there is also a reduction in the control-speed range $N_\infty - N_0$ when the spring preload is increased. Thus, if the control-point speed N_0 can be any arbitrarily chosen value, a high preload will increase the slope of the governor characteristic and will improve stability.

In practice, the control-point speed is usually a specified value (or range of values). To increase the slope of the governor characteristic, i.e., to obtain improved stability, with N_0 at a fixed value, there is the possibility of lowering the stability limit $N_\infty = \sqrt{k}$, e g., by using a softer spring.

The slope of the characteristic at the control point is

$$x' = (\mathrm{d}x/\mathrm{d}N)_{N_0} = 2N_0s_0^2/[k(s_0 - e)] \quad .. \quad (4)$$

To keep N_0 unchanged, eq (2) indicates that a reduction in k (i.e., of the stiffness K) must be compensated by an increase in preload. This gives $ke = (k - \triangle k)(e + \triangle e)$ = constant, from which the increment $\triangle e$ of the preload setting can be determined. Let $x^{*\prime}$ be the slope at N_0 of the new characteristic. By means of eq. (4), it can be shown that the ratio $x^{*\prime}/x'$ is greater than unity, i.e., that the slope is increased with a softer spring. It can be verified by numerical evaluations of concrete cases that the improvement is considerable and that the control range ($\triangle N = N_\infty - N_0$) is cut down more rapidly than the spring stiffness.

Another possibility of increasing the effectiveness of governing is to modify the weights in such a manner that the distance s_0 of the centre of gravity will be increased. This can be achieved, for instance, by inserting

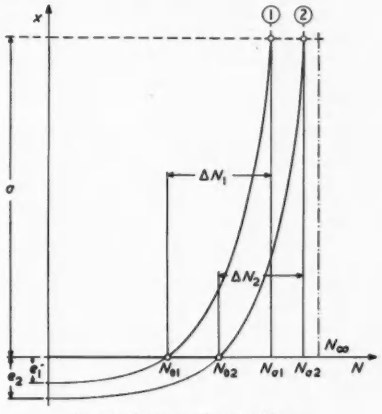


Fig. 2. **Governor characteristics.**

wo lead plugs in holes drilled in the outer section of each weight. In this case also, the control-point speed N_0 is kept constant by a corresponding increase in preload setting (the condition being $N_0^2 = e/(ms_0C) = (e + \triangle e)/[(m + \triangle m)(s + \triangle s_0)C]$ = constant, where $C = 0{\cdot}01096$). It can be shown that, with lead plugs in the iron weights, the value of $N_\infty = \sqrt{k}$ is reduced ; but the reduction in the control range and the increase of the slope are so slight that it can be concluded that this second possibility is hardly useful.

A third possibility is that of using a heavier metal for the weights, without altering their design or their centre of gravity. For instance, the mass of the weights can be increased by about 11% if they are made of brass, giving an appreciable improvement in the steepness of the characteristic, although not as great as that obtainable with a reduced spring stiffness.

In the above analysis, definitions have been given for the limiting speed N, the control-point speed N_0 and the theoretical control range $\triangle N = N_\infty - N_0$. In addition, it is useful to define the "stable operating range" N_{st}, which extends from a point somewhere near N_∞, where the springs are not coilbound, to a point on the left of N_0 on the flat straight portion of the characteristic, where Ke is greater than the centrifugal force. Within this range, sufficient fuel is available for the engine to overcome internal resistances. Finally, the concept of a "regulation parameter" R is defined as

$$R = N_0/N_\infty = \sqrt{e/s_0} \quad .. \quad .. \quad .. \quad (5).$$

With this parameter, it can be seen that improved governing (i.e., increased values of R) can be obtained by increasing the preload setting e, but this moves the control point N_0 towards the right. If N_0 is to be kept constant, the only factor that can be varied is N_∞ ; in other words, it is necessary to reduce the value of k. It may be noted that $R = 0$ when e is zero ; also that, for an "ideal governor", $R = 1$.

For a comparative assessment, it is of interest to consider the R-values which have been obtained in calculations for a particular fuel-pump governor, with $K = 3300$ kg/m and $m = 0{\cdot}0724$ kg sec²/m, giving $N_\infty = 2040$ and $N_0 = 1053$ rpm camshaft speed, so that, in the initial arrangement, $R = 1053/2040 = 0{\cdot}516$. It was found that, when e was increased by 20%, N_0 being increased correspondingly, $R = 0{\cdot}565$. With N_0 kept constant and with an increase in K of 24·2% and an increase in e of 32%, $R = 0{\cdot}593$. With N_0 constant and using lead plugs, with an increase in e of 1·7%, $R = 0{\cdot}519$. With N_0 constant and using brass weights, representing an increase in m of 11·1% and an increase in e of 11·5%, $R = 0{\cdot}545$.

GERMANY

Dynamic Tooth Forces

By G. Niemann and H. Rettig. (From *VDI Zeitschrift*, Vol. 99, No. 3, January 21, 1957, pp. 89-96, and No. 4, February 1, 1957, pp. 131-137, 54 illustrations.)

(*Concluded from May issue*)

Effect of Speed (Pitch-Line Velocity) on Tooth-Deflection Amplitude

Fig. 10 shows as an example the variation of the maximum values of tooth-deflection amplitude A_{tot} of the oscillographic records with speed n for the test series with a tooth error $f_e = 40\mu$. It can be seen from this figure that

(1) The tooth deflection under static load corresponds to A_{tot} for $n = 0$.

(2) The curves through individual points (dotted lines) have a marked waviness, which leads to the conclusion that resonance with natural frequencies of the system occurs in the region of the maxima.

(3) The mean curves (solid lines) are almost level above $n \approx 200$ rpm, if the lowest curve (very low load) is ignored. The smaller the load, the greater is the ratio A_{tot}/A_{stat}, and the speed range in which A_{tot} still rises.

Effect of Speed on P, σ and K

In spite of the insignificant rise of the measured A_{tot}-curve in Fig. 10 above $n = 200$ rpm, the corresponding tooth loads P_{tot} and also the root stresses σ_{tot} and contact pressures K_{tot} increase with higher values of n. This difference arises largely from the fact that the tooth-deflection maxima (see Fig. 9d) are displaced towards the pitch-contact point with increasing speed, an additional reason being the frequency response of the measuring bridge. This means that the corresponding tooth load comes into action closer to the root of the measuring tooth ; an equal deflection at the pick-up then corresponds to a higher tooth load. Furthermore, the following points may be noted with regard to the speed effect :—

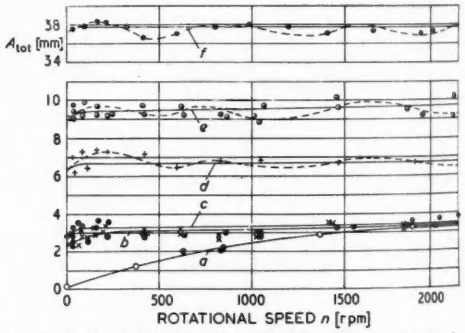


Fig. 10. Tooth-Deflection amplitude A_{tot} of the oscillographic records as a function of rotational speed n for various static loads P_{stat}, with normal gear mass and base-pitch error $f_e = 40\mu$.

(a) $P_{stat} = 1{\cdot}42$ kg/cm. (b) $P_{stat} = 34{\cdot}6$ kg/cm.
(c) $P_{stat} = 42{\cdot}2$ kg/cm. (d) $P_{stat} = 91{\cdot}1$ kg/cm.
(e) $P_{stat} = 135$ kg/cm. (f) $P_{stat} = 562$ kg/cm.

Test gears :—30 teeth ; 20-deg. pressure angle ; 3-mm module ; addendum modification factor = 0·2615.

(1) From Fig. 11 it can be seen that P_{tot}, σ_{tot}, and K_{tot} rise almost linearly with n (for $n > 200$ rpm). The rate of rise increases with increasing tooth error f_e and load P_{stat}.

(2) The determining ratios P_{tot}/P_{stat}, $\sigma_{tot}/\sigma_{stat}$, and K_{tot}/K_{stat} increase more slowly with speed for large values of P_{stat}.

(3) The increase in the ratios with increasing speed is highest for P_{tot}/P_{stat}, and lowest for $\sigma_{tot}/\sigma_{stat}$.

(4) For $n = 0$, $P_{tot} = P_{stat}$ and is independent of the tooth error f_e, whilst σ_{tot}, even for $n = 0$,

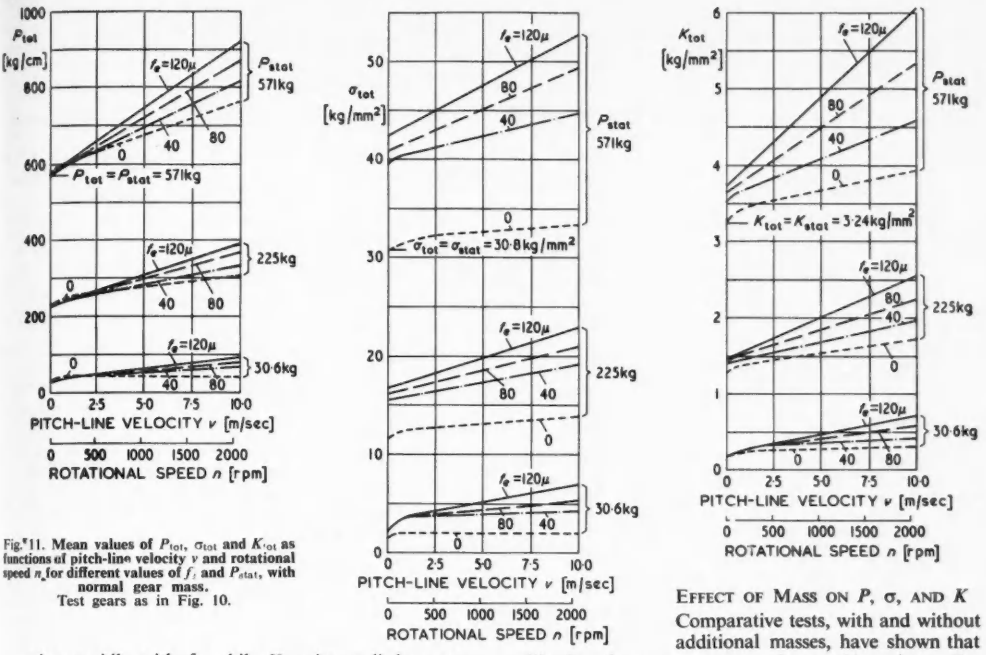


Fig. 11. Mean values of P_{tot}, σ_{tot} and K_{tot} as functions of pitch-line velocity v and rotational speed n for different values of f_e and P_{stat}, with normal gear mass.
Test gears as in Fig. 10.

rises rapidly with f_e, while K_{tot} rises a little more slowly than σ_{tot}. This relation checks with static measurements, in which it was found that a positive tooth error leads to early loading of the measuring tooth, thus increasing σ_{tot} and K_{tot}.

Effect of Tooth Error on P, σ, and K

Evaluation of Fig. 11 has established that

(1) Dynamic loads arise even for perfect gears ($f_e = 0$).
(2) With an increasing value of f_e (for $f_e > 20$ μ), P_{tot}/P_{stat}, $\sigma_{tot}/\sigma_{stat}$, and K_{tot}/K_{stat} also rise almost linearly. This rate of rise increases with increasing values of n and P_{stat}.
(3) With increasing values of f_e, the ratios P_{tot}/P_{stat}, $\sigma_{tot}/\sigma_{stat}$, and K_{tot}/K_{stat} rise much more slowly for large values of P_{stat}.
(4) The increase in the ratios with increasing values of f_e is greatest for $\sigma_{tot}/\sigma_{stat}$, and almost always least for P_{tot}/P_{stat}.

Effect of Load on P, σ, and K

The absolute values of P_{tot}, σ_{tot}, and K_{tot} have been found to rise almost linearly with P_{stat} (above about 50 kg/cm) and, as can be seen from Fig. 12, the ratios P_{tot}/P_{stat}, $\sigma_{tot}/\sigma_{stat}$, and K_{tot}/K_{stat} decrease considerably with increasing values of P_{stat} and approach lower limiting values, which depend on n and f_e. The reduction in the ratios is particularly great for large tooth errors; consequently, the influence of the dynamic incremental load is especially important for gears of low load capacity, i.e., for unhardened gears, for which the maximum flank load is the determining factor, and where especially the ratio K_{tot}/K_{stat} is quite large, even for small tooth errors. For example, with $P_{stat} = 30{\cdot}6$ kg/cm, $f_e < 5\mu$, and $n =$ 500 to 2000 rpm, $K_{tot}/K_{stat} \approx 1{\cdot}50$ to $1{\cdot}73$. In this example, $\sigma_{tot}/\sigma_{stat} = 1{\cdot}21$. With $f_e = 40\mu$, $K_{tot}/K_{stat} = 1{\cdot}88$ to 2·50, and $\sigma_{tot}/\sigma_{stat} = 2{\cdot}24$ to 2·58.

Effect of Mass on P, σ, and K

Comparative tests, with and without additional masses, have shown that

(1) In the range $v = 0$ to 11 m/sec, P_{tot} and K_{tot} increase in the same way as with a normal gear mass, but P_{dyn}, σ_{dyn}, and K_{dyn} are considerably greater, especially with low static loads and large tooth errors, where the increase is about 50%.
(2) The way in which the error affects P, σ, and K is the same as without additional masses.
(3) With larger additional masses, the values of P_{tot}, σ_{tot}, and K_{tot} mainly increase in the region of small static tooth loads, whilst, for values of P_{stat} greater than 300 kg/cm, the effect of the gear masses is no longer noticeable.

Order of Magnitude of Dynamic Tooth Loads

From tests without additional masses and with $f_e = 0$ to 80μ, $P_{stat} = 30{\cdot}6$ to 570 kg/cm, and $n =$ 500 to 2000 rpm, it has been found that P_{tot}/P_{stat} lies between 1·10 and 2·80, $\sigma_{tot}/\sigma_{stat}$ between 1·03 and 3·30, and K_{tot}/K_{stat} between 1·10 and 3·24. These ratios can therefore reach considerable magnitudes.

Check Tests

In order to ascertain how dynamic loads affect the strength of the tooth root in practice, a range of Wöhler curves was obtained from running tests on gear pairs of various base-pitch errors, materials, and modified tooth forms.

In a series of tests on grey cast-iron gears meshing with incorrect gears of hardened steel, the error was deliberately put on the hardened gear, so that it would decrease as little as possible during operation. The results showed that, for gears of relatively low load capacity, the maximum steady load that can be carried decreases very sharply with increasing tooth error f_e, i.e., with increasing dynamic load. Thus, for example, with $f_e = 160\mu$, the steady load is only 10% of that obtainable

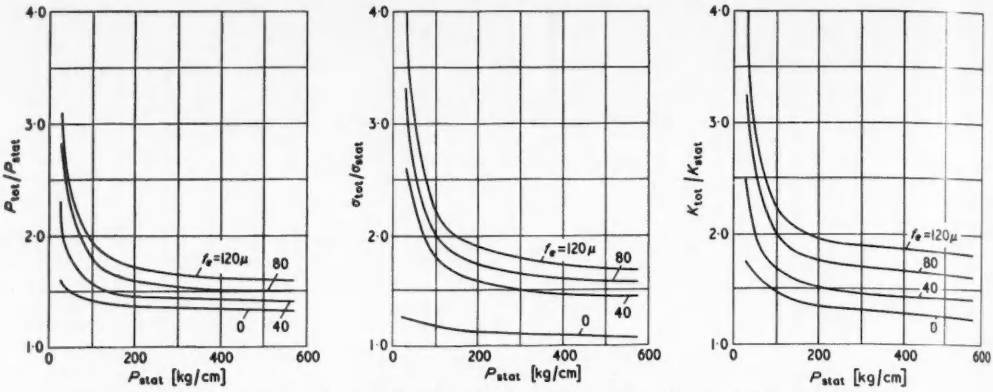


Fig. 12. Ratios P_{tot}/P_{stat}, $\sigma_{tot}/\sigma_{stat}$, and K_{tot}/K_{stat} as functions of P_{stat} for n = 2000 rpm and different values of f_e.

with $f_e < 10\mu$. It was also established that incorrect tooth flanks which have become worn down at the root and tip (see Fig. 8) give rise to an appreciably smaller dynamic-load increment than undiminished tooth errors. These results are in agreement with the conclusions derived from dynamic tooth-deflection measurements. For small tooth loads (unhardened gears) and for small errors, the dynamic incremental load can reach very high values, compared with the static load.

In tests with both gears of hardened steel, all the test points for large and small tooth errors were found to lie in the same scatter band, while an increase in gear mass or in speed did not produce any noticeable change. From these tests it is concluded that, for heavily loaded (hardened) gears in the range of tests carried out ($0{\cdot}4 \times 10^6$ to 10×10^6 load reversals), the magnitude of the tooth error has only a slight effect on the tooth root strength. For the region beyond 10^7 cycles, additional tests would have to be instituted. The reasons for this are to be found, as confirmed by the dynamic tooth-deflection tests, in the reduction of $\sigma_{tot}/\sigma_{stat}$ with increasing values of P_{stat}, and also in the large reduction of tooth error with operating time, with heavy loads and large errors.

In tests on hardened gears, using tooth forms with a modified addendum, it was confirmed that, for high static loads (hardened gears), the change in dynamic incremental load was not of great importance. These results could also be regarded as a confirmation of the calculation method[12] for root fatigue strength σ_D (based on photoelastic tests), as this also leads to constant limiting values for σ_D for various tooth forms, in agreement with tests.

Tests on tooth forms with the same effective leverage e (Fig. 13) and different contact ratios were also effected, the addendum of the test gear being altered in steps in such a way that the theoretical contact ratios (ε = 1·80, 1·36, and 1·10), but not the leverage e, changed at the outer point of single tooth contact. It was found that the static values of the root fatigue strength σ_D from running tests on hardened gears (20-deg. unmodified tooth form, with reduced addendum) remained unchanged, with P_{stat} = 724 kg and σ_{Dstat} = 41·3 kg/mm².

Additional tests with different leverages e and contact ratios were also carried out, in which the addendum of three meshing gears was altered and consequently the leverage e of the test gears changed, as also was the contact ratio of the gear pairs. The results showed that, even under widely varying tooth conditions, the values of the root fatigue strength σ_D remained almost unchanged.

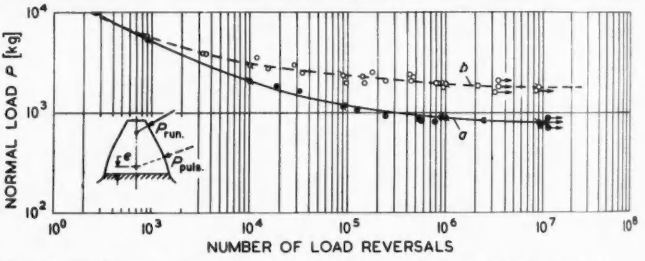


Fig. 13. Wöhler curves of root strength from running tests (curve *a*) and pulsator tests (curve *b*) at n = 1450 rpm.
Diagonal through points indicates calculated value ; all other points are test values.
Test gears :—Chromium-manganese case-hardened steel; 20-deg. nominal pressure angle; 3-mm module ; 10-mm face width; addendum modification factor of gear (39 teeth) = 1·00, and of pinion (19 teeth) = 0·75.

For purposes of comparison with the results of the above tests, a number of additional tests was also carried out on a pulsator machine. It was found that the curves of root strength obtained from these pulsator tests differed quite definitely from those from running tests, as shown in Fig. 13. Based on the tooth-deflection tests and numerous comparative tests on gears of different strengths, the following comments can be made :—

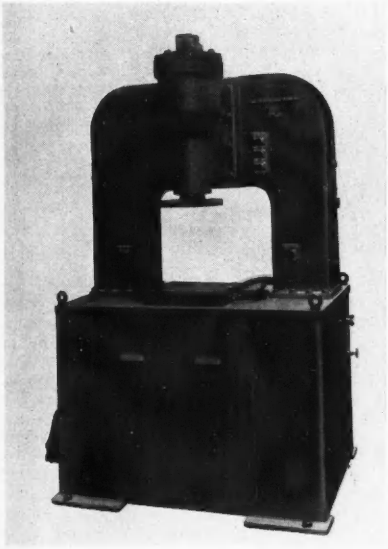
(1) The true contact ratio ε_w increases with load in running tests, with the result that the effective leverage e decreases at the decisive point of single tooth contact, whereas in pulsator tests it remains almost unchanged. Thus, the Wöhler curve from running tests (curve *a*) rises more sharply in the region of high loads (limited life) than the curve from the pulsator tests (curve *b*).

(2) The values of the root fatigue strength σ_D obtained from running tests on both hardened gears and on grey cast-iron gears are only 70 to 75% of the values obtained from pulsator tests, provided that tooth errors are less than 6μ. This difference, at least for hardened gears, cannot be accounted for entirely by the dynamic incremental load in the running tests. The cause must be sought, at least in part, in the different forms of the load cycles and in the different running conditions.

British Industrial Developments

Air-Operated Series Projection Welder

Where the mass production of resistance-welded components is required, series projection welding, where applicable, frequently is of advantage, compared with the more usual methods. In this connection, the Metrovick type APS60/24 air-operated series projection welder, announced by Metropolitan-Vickers Electrical Co. Ltd., of Trafford Park, Manchester, uses a bridge-type construction, with a gap of 24 in. between the vertical members, which support an 8½-in. bore cylinder developing a force of 4500 lb. Although the transformer is rated at only 60 kVA, the capabilities of this machine are equal to those of a conventional unit of much higher rating.



The thyratron timer for controlling the welding period, the ignitron contactor, and the transformer are located in the base of the frame. The standard machine can be fitted with welding jigs to suit particular requirements, e.g., for the simultaneous welding of two bosses to the inner surfaces of a compressor shell. Another application for which this machine can be used is the multiple welding of crossed wire joints and, within the area of the double lower platen, which measures 12 × 13 in., as many as 80 joints in 12-gauge mild-steel wire can be welded simultaneously.

Cold-Casting Filler Metal

Capable of being easily spread in paste form, and comprising a liquid constituent of thermosetting plastics and a powder containing 70% of finely divided metals, cured by catalytic action, " Metolux ", announced by M.G. Plastics Ltd., of Blackburn, Lancs., possesses all the desirable properties of metal in a conveniently spreadable form. As such, it is, among other applications, eminently suitable as a filler for blow-holes in castings, etc., and for building up defects in all metallic parts. It can also be used in certain cases for moulding small components for decorative or other purposes, provided that they are not subjected to heavy wear.

In use, the powder is mixed with the liquid to a stiff paste and applied to the material, leaving the surface sufficiently proud to allow for finishing. The operation must be carried out within 7 min, as rapid hardening takes place after this period, final setting occurring after about 30 min. A very high standard of adhesion is obtained, ensuring perfect feathering and a crack-free surface. The filler does not shrink or expand in itself, but is sufficiently flexible to move with the metal to which it is bonded. Furthermore, it is resistant to vibration, rust, petrol, oils, greases, and most chemicals, and can be machined, ground, filed, or drilled. It can also be painted, using normal techniques.

Scale-Removal Jelly

Designed for the removal of heavy scale from hot-rolled steel plates, girders, etc., " Waltergel " is the latest addition to the range of processing chemicals produced by The Walterisation Co. Ltd., of Purley Way, Croydon.

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Direct-Cooled Stator Windings for Alternators above 200 MW

A system of direct cooling of stator windings for alternators over 200 MW is announced by The General Electric Co. Ltd., of London, W.C.2. The object of the development has been to produce an efficient cooling arrangement, both mechanically and electrically sound, and with the minimum disturbance to well-established methods and practice in manufacture. Thus, water is used as the coolant, since its high thermal capacity and low viscosity permit the use of small ducts, with a flow through several conductors in series, thus reducing to a minimum the number of tapping points on the winding. As a result, in the stator casing there is a complete absence of unusual insulation features, the leakage surfaces are normal, and all pipework is metallic, with brazed joints. Since the temperature rise of the water and the copper can be predetermined with accuracy, satisfactory operation of the cooling circuit is assured. A machine embodying this method of stator cooling is in course of construction.

The cooling arrangement used in this system is illustrated diagrammatically, the stator windings being represented by thick lines, with each phase divided electrically into two parallel circuits, this being essential in units of the size under consideration. The water flows into both ends of each phase winding, with discharges at the centre point of each half-phase, thereby giving twelve coolant circuits in parallel through the stator. The terminal

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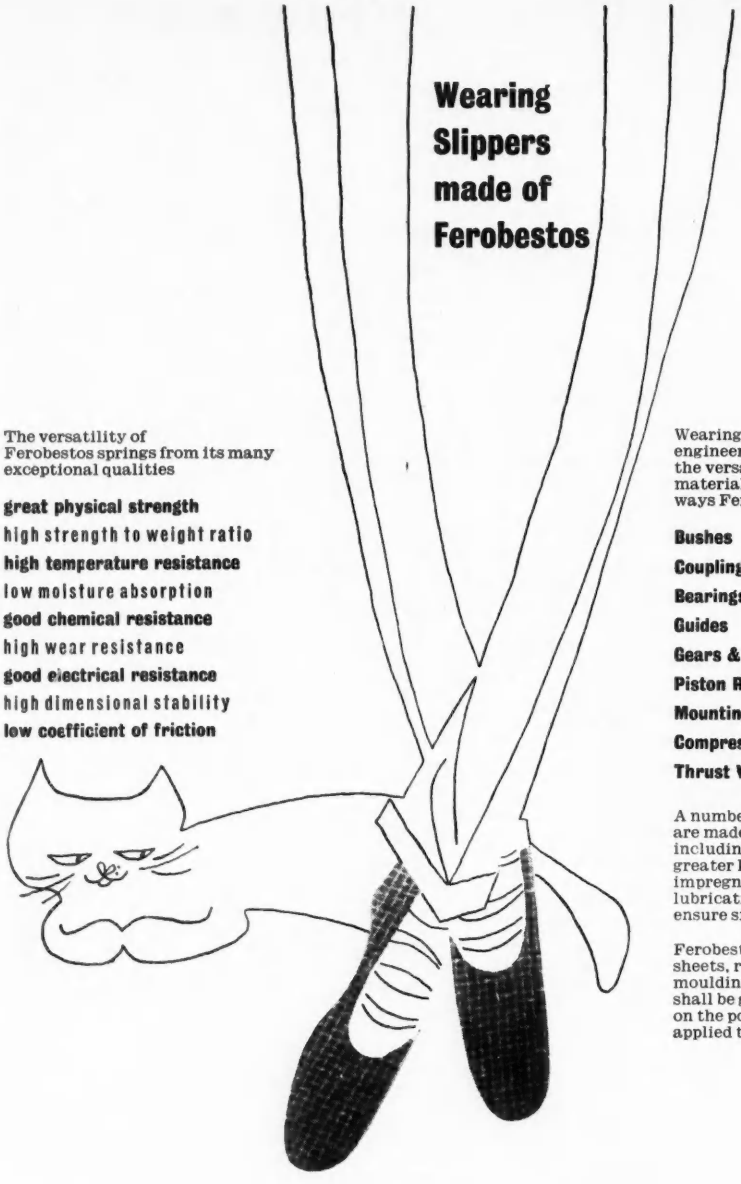

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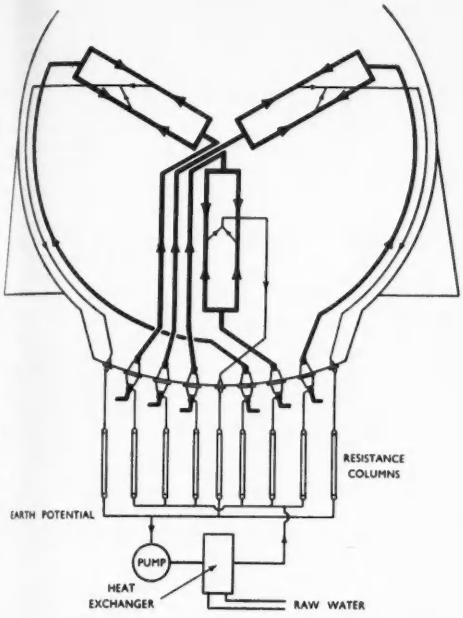


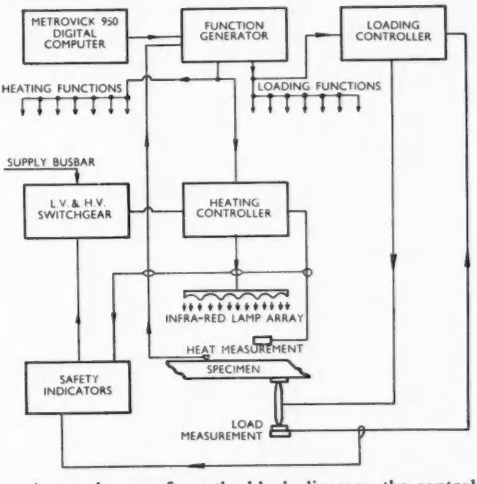
bushings have hollow stems through which the water passes into the casing, and the arrangement serves a double purpose, since the efficient cooling of the bushing by cold water greatly increases its current-carrying capacity and eliminates the possibility of any expansion strains. Three small bushings are also required for the warm water discharging from the windings, the connection between them and the mid-points of each phase being made with metal pipes insulated and supported in a similar manner to the electrical leads. In order to break down the voltage between the bushings and the earthed water-circulating pump and heat exchanger, resistance columns are provided at any convenient position in the foundation block. These are very simple and consist of straight tubes of insulating material of the appropriate length and diameter to contain a column of water of the required electrical resistance.

The G.E.C. water-cooled stator conductor consists of a number of thin-walled but robust copper tubes of reasonable depth, spaced symmetrically within the conductor, each tube being associated with two solid-copper strips placed above and below it, with only light insulation interposed. The water circulates through all tubes in parallel, the quantity through each being designed to absorb the loss both in the tube itself and in the associated strips. Since the heat is transferred from the strips to the tube over a wide area, the temperature gradient across the insulation is negligible, and all the copper in the conductor at any section is at substantially the same temperature. One of the chief advantages of this arrangement, compared with one using tubes alone, is the greater freedom it gives in the selection of the most suitable proportions for the various components, from the point of view of losses, hydraulic efficiency, and ease of handling.

It is well known that the stator conductors of all but the smallest alternators are subdivided to reduce eddy-current losses, the usual method being to use a number of horizontal laminations which are individually insulated and transposed. The method of transposition used by the G.E.C. is such that each strip is cyclically transferred from one position in the slot to an adjacent position when making the joints between conductors in the stator overhang, the interstrip insulation being continuous throughout the winding. This arrangement has the advantages of very simple interstrip insulation, uniform depth of copper, and the highest space factor in the slot, with good equalization of induced stray voltages in the stator overhang. The same principle is used for the direct-cooled winding but the design is slightly modified to eliminate the need for " jumpers " normally used to transfer the required number of strips from the top to the bottom of the stack of laminations when making the brazed joints to the next coil. At both ends of the stator the strips and tubes follow round from top to bottom in an orderly sequence, which is, of course, very desirable with a direct-cooled winding, to facilitate the jointing of the tubes.

Simulation System of Kinetic Heating in Aircraft and Guided-Missile Structures

Bearing in mind the vital importance of temperature distribution in the structure of modern aircraft and guided missiles, the Metropolitan-Vickers Electrical Co. Ltd., of Trafford Park, Manchester, has developed a complete system for simulating in the laboratory the conditions likely to be met during high-speed flight and for studying the deformations caused by the thermal stress, where unequal temperature distributions caused by changes in speed arise. The new system has been designed to provide a co-ordinated method of conducting kinetic heating tests, while retaining the flexibility necessary to meet the various requirements of different users. Provision has been made to supply either controlled alternating or direct current to cater for pilot schemes or full-size test projects. The power controllers may be either static or dynamic, all being available in types having the required response.



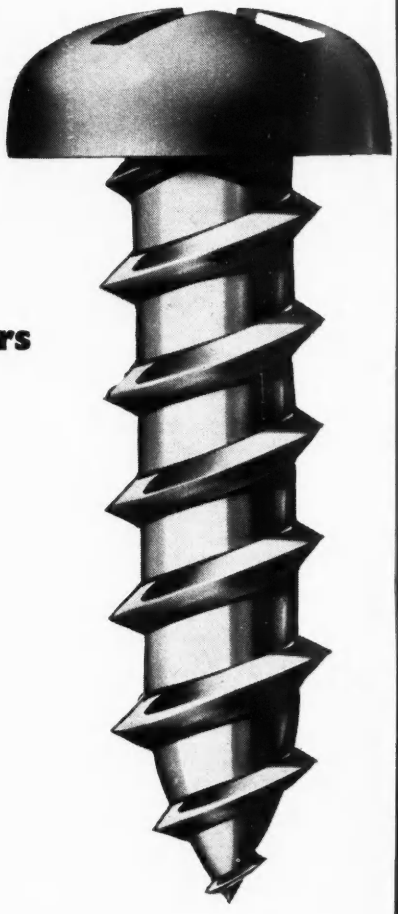
As can be seen from the block diagram, the control system for heating and stressing essentially comprises a " Metrovick 950 " digital computer, a function generator, and a number of loading and heating controllers (only one each of which, for simplicity, is shown in the diagram). The computer is used to produce the parameter programme on which the tests are to be conducted, and its output can be on five-hole punched paper tape, which is then fed to the function generator.

The simplified equation for heat flow into the structure gives

$$Q/A = h(T_w - t_s)$$

the more you think of it

the better it appears



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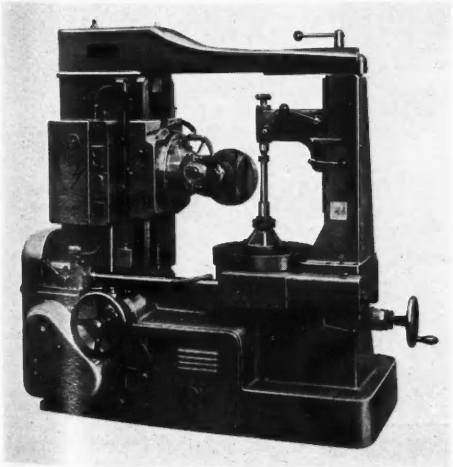
where Q/A is the computed heat perunit are a (required), h is the heat-transfer coefficient, T_w is the adiabatic wall temperature, and t_s is the specimen surface temperature. The function generator has been designed with a view to the addition of further terms in cases where a more rigorous solution is required. It has also been arranged to provide, where necessary, a programmed control of forces applied to the structure during the test in synchronism with the heating cycle. The heating and loading programmes can be produced either by a single function generator or, where the time scales differ, by separate function generators. Systems can also be arranged to control the specimen surface temperature t_s, instead of the heat flow Q, if so desired.

The outputs of the function generator are taken to the several heating and loading controllers, the output of the heating controller being taken to a bank of infra-red lamps, suitably located with respect to the structure specimen. The heat-measurement devices feed back signals to the function generator and heat controller to maintain the desired heat function. Similarly, the output of the loading controller is taken to the load actuator and load-measuring device, which applies the required force to the structure and provides a signal that is fed back to the load controller. For protection in the event of structural or system failure, circuits are included which can be arranged to shut down the heating and loading systems with minimum delay. High-voltage and low-voltage switchgear can also be supplied to suit requirements.

High-Capacity Automatic Gear-Hobbing Machines

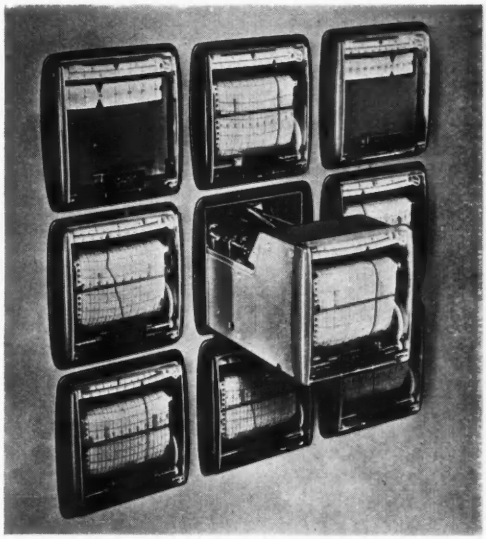
Manufactured in Sweden by Älmhults Bruk Aktiebolag, two new high-capacity automatic gear-hobbing machines, types KF-1 (illustrated) and KF-2, are announced by the Mortimer Engineering Co., of Harlesden, London, N.W.10. These machines, with respective maximum gear-hobbing diameters of 23½ and 47 in., are exceptionally robust, accurate, and easy to operate, and have been designed for the economic hobbing of gears both for one-off and mass production.



On cylindrical blanks, these machines can, by means of a work hob on the generating principle, be used to hob helical and spur gears, sprockets, ratchet-wheels, worm-wheels, splines, etc. In addition, they can be equipped with a tangential-feed hob head for cutting worm-wheels on the single-tooth principle and worms using cutting wheels as tools.

"Pull-Out" Recorder/Controller

Recently introduced to the range of "Consotrol" instruments produced by Foxboro-Yoxall Limited, of Merton, London, S.W.19, the M/54 recorder/controller is similar to, and performs the same functions as, the M/53 series, but incorporates an extra facility whereby the recorder, complete with the controller (when mounted at the rear of the recorder) can be removed completely from the panel whilst the transfer switch is in the "Seal" position. The entire unit slides into a sheet-steel housing



which is secured to the panel and to which are attached all the external connections. These, electric and pneumatic, are of the plug-in type and, when the unit is withdrawn from the panel, a series of air check valves automatically closes, preventing loss of pressure in the impulse lines. The housing completely encloses the M/54 mechanism, which is thus air-purged, protecting the instrument from harmful atmospheres and increasing the safety of operation where an electrical chart drive is used.

The complete chart-drive assembly may be removed from the unit without affecting the process operation. The assembly is removable as a unit, and a mercury switch automatically breaks the chart-drive circuit. Where a pneumatic chart drive is used, a check valve performs the same function. A full 4-in. width chart is used, and the panel area required is only 6 × 6 in., so that M/54 instruments can be mounted on centre-lines as close as 7 in., without impairing accessibility.

When the M/54 is used as a recorder/controller, the M/58 "Consotrol" controller may be integrally mounted with the recorder, and the ease with which the instrument can be withdrawn from the panel simplifies the adjustment procedure. All controller settings are easily accessible from the front of the panel. Rear access to the adjustments is provided by means of a hinged, plastic window at the rear of the M/54 case. The pull-out feature permits a complete instrument to be replaced without disturbing the process. Since the control system can be transferred from automatic to manual control, the M/54/58 can be removed from the panel and replaced by a service spare unit, on which the setting index and controller settings have been adjusted to agree with the values already obtained. Bumpless transfer, using the transfer switch, ensures that the system is brought back into completely automatic operation a few minutes after the change-over operation is commenced.

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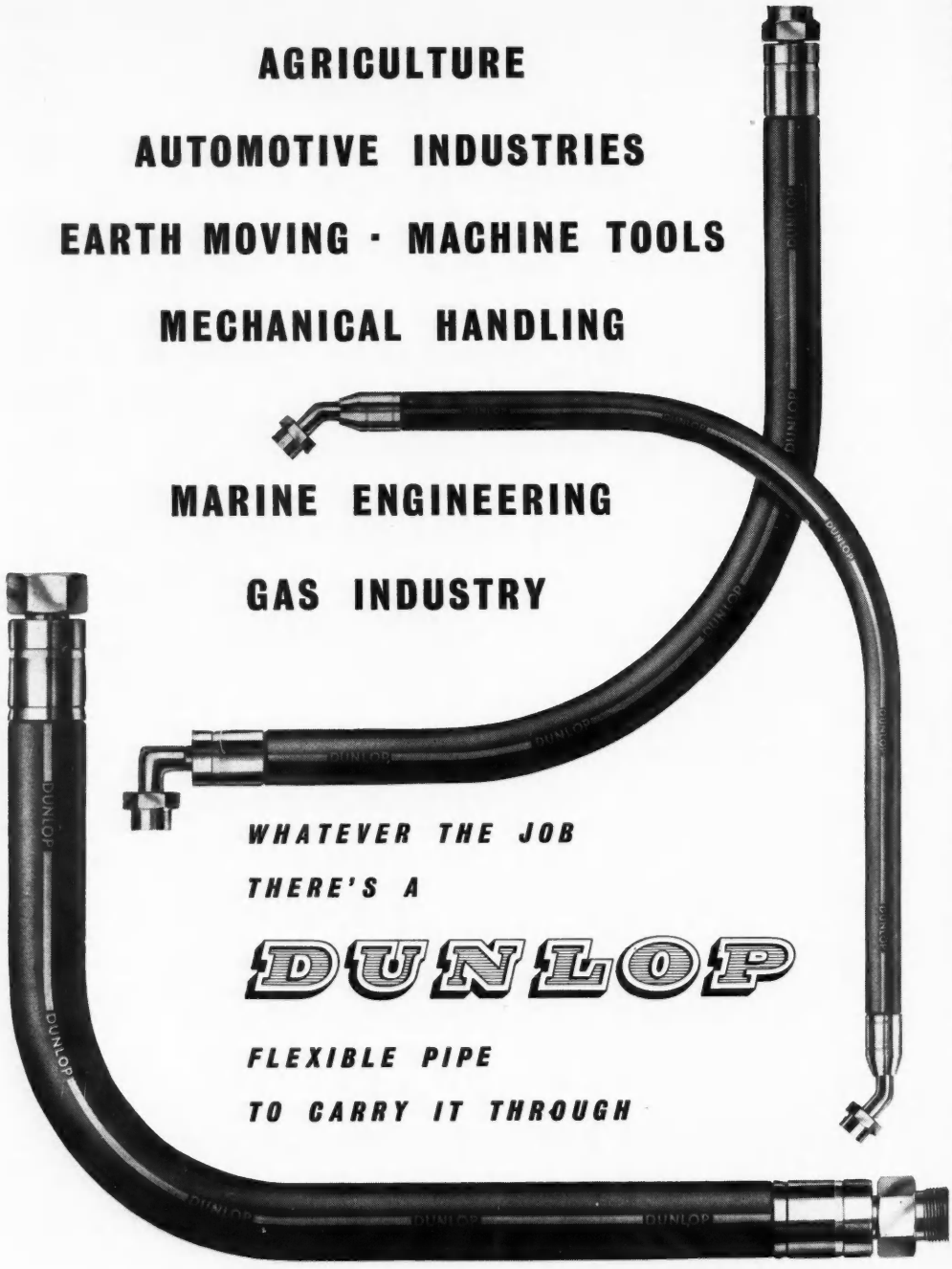

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ADDITIONAL ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

Stress and Deflection Studies of Pipeline Expansion Bellows.

By C. E. TURNER and H. FORD. (From an advance copy of a paper submitted to *The Institution of Mechanical Engineers, London*, May 1957, 19 pages, 26 illustrations.)

THE use of high-temperature fluids has posed many problems in engineering, not the least of which are those arising from the expansion of a heated pipeline. A form of corrugated pipe, known generally as an expansion bellows, has been used as one solution to this problem. Such a bellows, the geometry of which may vary considerably from one application to another, can either be incorporated in the pipeline—particularly for low-working pressures—or used as a flexible anchorage through which the pipe passes. The movements of the bellows are usually either angular (i.e., a hinge joint) or axial. A specific example of the axial movements of a flexible anchor-point can be found in ship-board use where a pipe passes through a bulkhead. The bellows provides both a flexible anchorage and a watertight seal ; no pressure loading arises in this case. This paper presents a theoretical solution, together with an experimental study of the axial compression of certain bellows, mainly of the corrugated-pipe type, used in the pressureless state.

An approximate theory has been developed for the compression of a bellows, the convolutions of which have a cross-section formed by circular arcs subtending any semi-angle α. Numerical results have been calculated for the two cases $\alpha = \pi/2$ and $\alpha = 3\pi/4$. Experiments have been carried out on six bellows, four of the corrugated-pipe type ($\alpha = \pi/2$), one S-type ($\alpha = 3\pi/4$), and one flat-plate type which was not analysed theoretically. Deflection and resistance strain-gauge readings were taken on each bellows, and reasonable agreement was found between the theoretical and experimental results for the five bellows in which these could be compared.

It has been found that, for certain design conditions, optimum relationships exist between bore, wall thickness, and radius of convolution for maximum flexibility. Stress-intensification and flexibility factors have been calculated to cover a range of bellows proportions.

A Comparison of Refrigerants in Vapour-Compression Cycles over an Extended Temperature Range.

By J. P. BARGER, W. M. ROHSENOW, and K. M. TREADWELL. (From *Transactions of the ASME, U.S.A.*, Vol. 79, No. 3, April 1957, pp. 681–685, 7 illustrations.)

THE nature of the problem of cooling supersonic aircraft is now becoming clear. Heat will be transferred from sources inside the aircraft at various prescribed temperatures to a sink outside the aircraft at a higher temperature. The stagnation temperature is the temperature of the sink to which the heat must be rejected from the aircraft. Even if ram air-expansion turbines, not popular with aircraft designers because of their large size and weight, can be employed, the sink temperatures at Mach numbers greater than 3 are still much higher than those which the refrigeration engineer normally encounters.

The heat-source temperatures depend on the function of the aircraft. In general, the entire aircraft will not be cooled ; parts of it will be allowed to approach stagnation temperature, but the temperature of three main areas of the aircraft will be reduced, i.e., (1) certain parts of future hydrocarbon-burning engines will require cooling to keep metal temperatures below about 1540°F ; (2) the temperature of electronic equipment of current design will be kept below approximately 240°F ; and (3) the cabins of manned aircraft will be maintained at temperatures between 50 and 70°F. At any given Mach number the temperature range between source and sink can be determined by comparing one of the source temperatures with the stagnation temperature corresponding to the Mach number.

It will be appreciated that an important engineering problem involves the provision of apparatus to meet the cooling requirements of supersonic aircraft. Because the time is already passing when simple air cycles can render adequate performance, owing to their inability to extend over a sufficient temperature range, vapour compression cycles are being considered. However, the maximum temperature range over which a single refrigerant can operate successfully in a vapour cycle is also relatively limited. Since the total required temperature range increases rapidly with Mach number, series combination or "stacking" of vapour cycles, leading to binary and higher-order cycles, may become necessary. This paper gives methods of selecting the refrigerant or series combination of refrigerants exhibiting optimum thermodynamic performance when employed in vapour cycles operating between arbitrarily selected temperatures.

Observation of Dislocation Motions in Crystals.

By J. J. GILMAN and W. G. JOHNSTON. (From *Metal Progress, U.S.A.*, Vol. 71, No. 3, March 1957, pp. 76–77, 3 illustrations.)

As described in this brief article, it is now possible to observe dislocation motions in crystals. Although the three illustrations included (showing the glide of dislocations during annealing and plastic bending and of an isolated dislocation) are of non-metallic crystals of lithium fluoride, they show by analogy how dislocations probably move in metal crystals.

Specimens of lithium fluoride prepared by cleavage have flat mirror-like surfaces. When these surfaces are etched in 1 part of concentrated HF, 1 part glacial acetic acid, and a pinch of FeF_3, they are not attacked uniformly. Almost no attack occurs, except where a line of dislocations intersects the surface. At these dislocation sites, small pits grow with a pyramidal shape. The etching attack is so sensitive to the presence of dislocations that, if the dislocation moves away from the bottom of the etch-pit, the pit immediately stops getting any deeper, although it continues to grow sideways. Thus, it becomes " flat-bottomed " or truncated and can be distinguished easily from the pointed pits at places where dislocations are still present. Thus, the " before " and " after " positions of dislocations which have moved can be seen.

Dislocations can move in two ways. They can " glide " easily if a shear stress is applied to them, and at somewhat elevated temperatures they can also " climb " when vacancies diffuse towards or away from them. Both of these processes are illustrated in the photographs, which were made with a metallograph using oblique reflected light. Transmitted light is not suitable for observing the pits. For work at high magnifications, the contrast can be increased by evaporating a thin film of silver onto an etched surface.

There is considerable evidence that the pits represent dislocations. They are located in subgrain boundaries in as-cast crystals. They are found arrayed neatly along glide planes in deformed crystals. When a deformed

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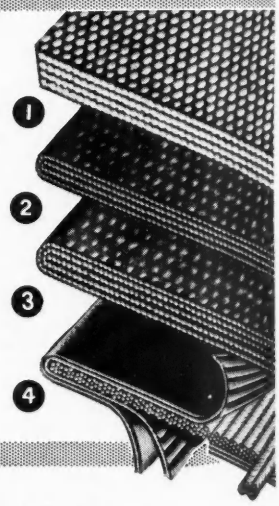

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crystal is annealed they disappear. At all events, the pits behave just as would be expected of dislocations. Furthermore, a dislocation is defined as a " line imperfection " in a crystal, and, by etching lithium fluoride crystals very deeply, it is found that the bottoms of etching pits follow definite lines going through the crystals and hence conforming with the definition of a dislocation.

Theoretical Studies and Laboratory Techniques in the Evaluation of Sea-Water Corrosion Testing.
By F. L. LaQue. (From *Corrosion, U.S.A.*, Vol. 13, No. 5, May 1957, pp. 33–44, 26 illustrations.)

Much has been learned about the behaviour of metals in sea water through laboratory investigations. Such tests are particularly useful in studying the corrosion processes by which metals are attacked and the reasons for and extent of the superiority of one metal over another, as it might be used in some practical application. Controlled tests are especially useful in appraising the merits of new or modified alloys and in screening a number of possible compositions by tests on small specimens, so that the large-scale trials required for final evaluation can be restricted to those alloys showing the most promise in laboratory studies. This paper describes the nature of some of these laboratory tests and outlines the advantages and disadvantages of each, with some notes on precautions to be taken in planning tests and interpreting their results.

Sodium chloride solutions offer certain advantages in laboratory testing, but the results in many cases are not the same as those which would be obtained in the more complex natural sea water. The composition of a satisfactory synthetic sea-water solution is given, together with data showing the correlation between corrosion rates of copper-alloy specimens in natural sea water and in synthetic sea water. The results of comparative spray tests, using natural sea water and synthetic sea water, are also reported (for cold-rolled steel and zinc).

The apparatus used in making simple immersion tests is described, the effects of high velocity and turbulence are discussed in some detail, and the results of the tests are reported for a number of metals. The results of jet impingement tests are also given. In addition, a description is given of the apparatus used in making corrosion-fatigue tests on metal specimens exposed to sea water, and discussions on crevice corrosion, galvanic potentials and polarization, and cavitation erosion are included.

Ultrasonic Machining.
By E. A. Neppiras and R. D. Foskett. (From *Philips Technical Review, Holland*, Vol. 18, 1956/57, No. 11, pp. 325–334, and No. 12, pp. 368–379, 24 illustrations.)

The technique of using high-frequency mechanical vibrations for machining brittle materials has assumed considerable importance in recent years, and, based on the growing interest in the ultrasonic machining technique, this article is intended as an accurate assessment of the potentialities of this method.

An introduction to the technique of drilling by ultrasonic vibrations is given. The actual cutting action consists of a continuous chipping of the work by abrasive particles in suspension, fed between tool and work. A vibration frequency in the ultrasonic region (about 20 kcs) is used, both for silent operation and in order to obtain reasonable cutting rates. Piezomagnetic (magnetostrictive) transducers are particularly well suited for generating the vibrations. Two types of laminated metal transducers are described, i.e., a bar-type and window-type, used in two ultrasonic drills developed in England by the Mullard Research Laboratories, the former in a small 50-W drill and the latter in a larger water-cooled 2-kW model. The associated velocity transformers are also described, these being resonant stubs, fixed to the transducer and holding the tool, which serve to amplify the vibrations at the tool face. Ultrasonic drilling can be applied only to hard, relatively brittle materials. Among the applications mentioned are cutting, the drilling of holes of any cross-section, and the production of bas-reliefs in such materials as glass, tool steel, sintered carbides, ceramics, and precious stones. Using a mechanical transmission line of flexible wire it is possible to drill holes in otherwise inaccessible places ; even curved holes can be bored. Other applications are the manufacture of blanking dies and wire-drawing dies and of precious-stone bearings for watches and instruments. The use of ultrasonic machining in transistor manufacture and in dental drilling is also mentioned.

Cutting speeds, accuracy, and surface finish are dealt with in some detail. The various factors affecting cutting speeds are considered with the help of graphs and tables. Cutting speeds increase almost linearly with static load until an optimum value (dependent on vibration amplitude and frequency) is reached. For tools of small area the adjustment to optimum load is critical. Within the range of usable amplitudes and at ultrasonic frequencies, cutting rates increase rather more than proportionally with amplitude and rather less than proportionally with frequency. Cutting speeds are higher with coarser abrasives up to a grain size limited by the vibration amplitude. Considerable data are given of actual cutting speeds in various metals and brittle materials (glass, ceramics, minerals, etc.), using different abrasives of various grain sizes. Cutting speeds vary from a few thousandths of an inch to some tenths of an inch per minute, according to the work material, abrasive, tool material, etc. Tool wear is also discussed briefly, together with the effect on accuracy of abrasive grain size and mechanical precision of the drill mounting. Two examples are given illustrating the accuracy obtainable (of the order of $\pm$ 0·0005 in.). Surface finish and its relation to abrasive grain size and vibration amplitude are also discussed.

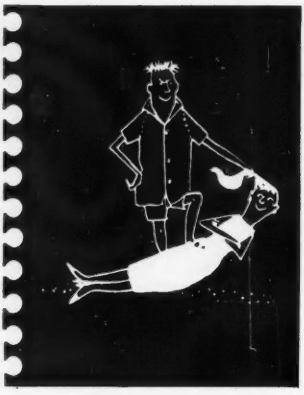
The Ductility of Tungsten-Arc Welds in Molybdenum.
By N. E. Weare, R. E. Monroe, and D. C. Martin. (From *The Welding Journal, U.S.A.*, Vol. 36, No. 6, June 1957, pp. 291s–300s, 3 illustrations.)

Molybdenum has many properties which would make it a useful structural material for many high-temperature uses. However, the major drawback to the use of molybdenum structures in non-oxidizing environments is low weld bend ductility between room temperature and about 400°F.

The objectives of the study reported in this paper were (1) to study some of the variables encountered in welding molybdenum, (2) to study the possibility of welding molybdenum outside a dry box, and (3) to determine if welds of consistent ductility could be made in different lots of arc-cast carbon-deoxidized molybdenum. Good bend ductility at 25° and 75°C was found for welds made using the best procedures established for dry-box and conventional welding.

Welds were made in the dry box to study the effects of cleaning methods, initial sheet microstructure, and two levels of atmosphere purity. Only atmosphere purity had a significant effect on weld ductility. Five methods of shielding welds made in air were studied, and the effects of type of inert gas used, gas flow, and nozzle size were determined with standard inert-gas shielding cups. A standard shielding cup and a leading-trailing shield unit have produced the best welds made in air to date. Other promising shielding methods are still being studied. The extent of tungsten contamination of molybdenum welds during tungsten-arc welding were also studied. About 3% tungsten in a molybdenum weld lowered weldment ductility. Weld porosity was associated with oxides or gas trapped along sheet edges.

Welds were made in six heats of arc-cast molybdenum with about the same carbon content, both in air and in the dry box for a consistency run. A difference in bend ductility was found between welds in the different heats.



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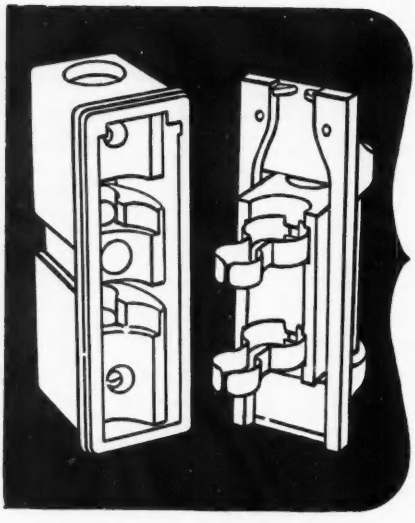

The Two-In-One Unit

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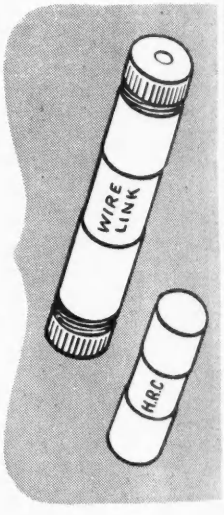
One of the outstanding features which have made the new "H" Type Slydlok Fuse universally popular is its ability to accept either Rewirable Fuselinks or H.R.C. Cartridge Fuses without any modification to the base or holder, a simple and economical arrangement which saves time, trouble and tempers.



Further advantages of the "H" Type and details of other fuses in the Wilcox range are given in our latest Catalogue, a copy of which will be sent to you on request.

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New Materials, Processes, and Equipment

RIBBON-WIRE PACKAGING TOOL

Developed from their power-tie oval-wire tool, the new ribbon-wire hand-packaging tool announced by Fredk. A. Power & Sons (Box Strapping) Ltd., of Birmingham, can be used on all types of packages, wooden boxes, cartons, etc. with absolute security and without the risk of damage occasioned by cutting into the package.

The tool is intended for the use of ribbon wire approximately 0·28 in. in width by 0·025 in. in thickness, i.e., equivalent in strength to $\frac{5}{16}$-in. steel strapping but cheaper. The twisted joint formed by the tool eliminates the need and additional cost of a separate seal. Unlimited tension can be applied easily and quickly, giving a joint of exceptionally high efficiency. The wire is supplied in a galvanized and rust-proofed finish.

In addition to its primary use for packaging, the new tool is also of value in facilitating the easy handling of materials in works, e.g., for handling raw materials or partly processed units for easy transport.

UNIVERSAL LAPPING MACHINE

Designed for the lapping of flat components up to $1\frac{1}{2}$ in. thick by 3 in. square, or cylindrical work up to $1\frac{1}{2}$ in. diameter by 3 in. in length (or 4 in. with oversize laps), the new " Unilap " universal lapping machine announced by Newall Group Sales Ltd., of Old Fletton, Peterborough, is a robust, compactly designed unit operated by push-button controls and capable of working to close tolerances. As such, it is of special interest to makers of small tools for the economical production of cylindrical gauges, faces of gauging blocks, references pieces for micrometers, comparators, etc.

The mechanical system follows the established Newall pattern of having superimposed laps, between which is a work-holder with a planetary drive. Stationary and floating, the upper lap incorporates hydraulic cushioning to prevent damage to components when it is lowered to the lapping position. The speed of the lower lap is 60 rpm, the work-holder being driven at a reduced speed, in order to ensure the same amount of stock removal simultaneously by both laps and to impart equal work to each lapping member. The lower lap and work-holding mechanism are driven by a 1-hp, 715-rpm motor through a bronze worm-wheel by a steel worm. This mechanism is completely enclosed in an oil sump, which also supplies automatic lubrication to all moving parts of the machine.

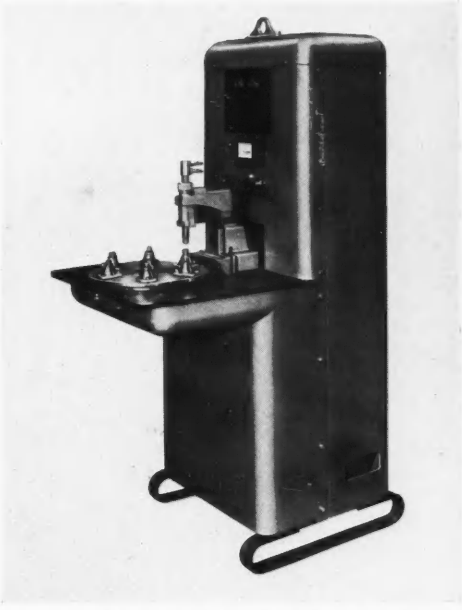
Fully universal, the " Unilap ", which is the smallest of three similar-purpose machines manufactured by the company, has almost unlimited applications within its capacity for the production of a wide variety of components employed in the manufacture of internal combustion engines ; it is also eminently suitable for lapping to high precision limits the ends and diameters of roller bearings, the sides of ball races, and thrust bearings.

INDEXING RESISTANCE-WELDING MACHINE

A new indexing resistance-welding machine, designed and manufactured by Hirst Electronic Ltd., of Crawley, Sussex, is intended for use in automatic production lines where spot or projection welding of non-ferrous and ferrous components is required. As such, it is ideally suitable for welding small components for electrical switchgear, such as silver contacts to brass and copper backing strips.

The motive power for revolving the table and depressing the welding head is compressed air, and outlets are available for operating loading and unloading devices. The table illustrated has six positions and delivers one component every four seconds, but this specification can be altered at will. Both welding electrodes and indexing table are water-cooled.

The welding head, though air-operated, retains low inertia properties, making the machine particularly suitable for ultra-short welding times down to 5 millisec, which may be required for non-ferrous welding. The timer for the welding current is synchronously controlled between $\frac{1}{2}$ and 9 cycles and is switched by an ignitron contactor. The maximum electrical capacity of the machine is 75 kVA, but smaller transformers can be supplied to order.



A monitoring device for ensuring that every weld receives an amount of current within certain acceptable limits is under development and will shortly be available for use in both portable and built-in form. With this device, any welds falling outside the prescribed limits, owing to fluctuations in the surface condition of components, mains fluctuation, etc., will be rejected.

CERAMIC MATERIAL FOR HIGH-TEMPERATURE HERMETIC SEALS

A new ceramic material for hermetic-seal applications at high temperatures is now available from the Consolidated Electrodynamics Corporation, of Pasadena, Cal. Known as " Ceramicite ", the compound bonds physically to most stainless steels and has a similar thermal expansion. Furthermore, seals embodying " Ceramicite " have exceptional ability to withstand severe physical and thermal shocks without leaks or cracking. The heating required by the sealing process causes unique changes in the linear-expansion properties of the materials, resulting in a common coefficient permitting crystalline migration between interfaces of the materials involved during the cooling cycle. Thus, an

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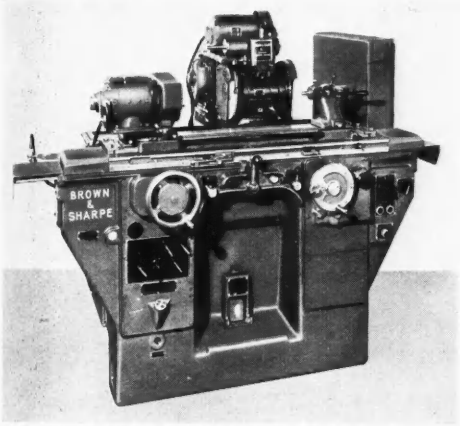
actual molecular bond exists between the metal and the compound, assuring a highly reliable, leak-tight seal with excellent electrical characteristics in continuous operation at temperatures up to 700°F.

"Ceramicite" is available in two forms, i.e., as preformed pieces, ready for firing by the user in fabrication of seal components, and as completed seal components, such as single- and multiple-pin terminals and headers. Typical uses include relay bases, transformer bases, transistor bases, diode closures, and other such components which require hermetic sealing.

UNIVERSAL GRINDING MACHINE WITH REMOVABLE EXTERNAL GRINDING SPINDLE

A new universal grinding machine, in which a special design of wheel-spindle head permits either the external grinding spindle or the internal grinding spindle to be used in any desired horizontal position, is announced by the Brown & Sharpe Manufacturing Co., of Providence, Rhode Island. Accurate compound settings can be made for combination straight and taper, or double-taper grinding (both external and internal) with only one set-up. A feature of this wheel-spindle head is a universal turret, on which the upper members of the head rest, and which can be turned and locked in any one of four angular positions. Turning the turret gives a 4-in. horizontal movement of the grinding spindle, either parallel or at right angles to the wheel-slide, extending the range of usefulness.

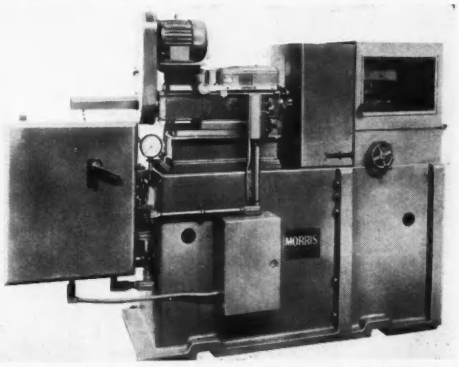


Spindle speeds of 2500, 3000, and 3600 rpm, together with work speeds from 60 to 600 rpm permit the grinding wheels to operate at the most efficient surface speed on an extremely wide range of work diameters for optimum surface finish and stock removal. The internal grinding spindle is hinge-mounted at the front of the wheel-spindle unit, where it can be instantly moved into operating position and rigidly clamped. Interchangeable internal grinding-spindle units, with anti-friction bearings and sealed lubrication, for speeds of 16,500, 24,000, or 35,000 rpm, are available, together, with a wide range of wheel arbors for each of the internal grinding spindles.

AUTOMATIC GUN-DRILLING MACHINES

Specially developed for increasing efficiency in many phases of gun-drilling operations, the new one-way and two-way models of automatic gun-drilling machines announced by The Morris Machine Tool Company, of Cincinnati, Ohio, provide infinitely variable spindle feeds and speeds, adjustable automatic load control, and a completely co-ordinated high-pressure coolant and chip-removal system. These machines are especially efficient for drilling deep holes of $\frac{1}{8}$ in. diameter and up. They are also suitable for shallow holes where straightness, size, and finish are crucial, and for jobs where speed necessitates a high rate of metal removal, and finish is not critical. Many sizes of gun drills, with variations in diameter and length, can be accommodated.

Spindle speeds up to 8000 rpm are obtained with a simple adjustment. Feeds up to 40 ipm are possible, with an infinitely adjustable feed stroke, speeds and feeds being completely independent of one another.



The adjustable load control senses conditions that might endanger the operation, protecting the machine against breakage of gun drills or workpieces. It automatically detects dull drills, chipped drills, hard spots in the material, or insufficient coolant, instantaneously reversing the quill.

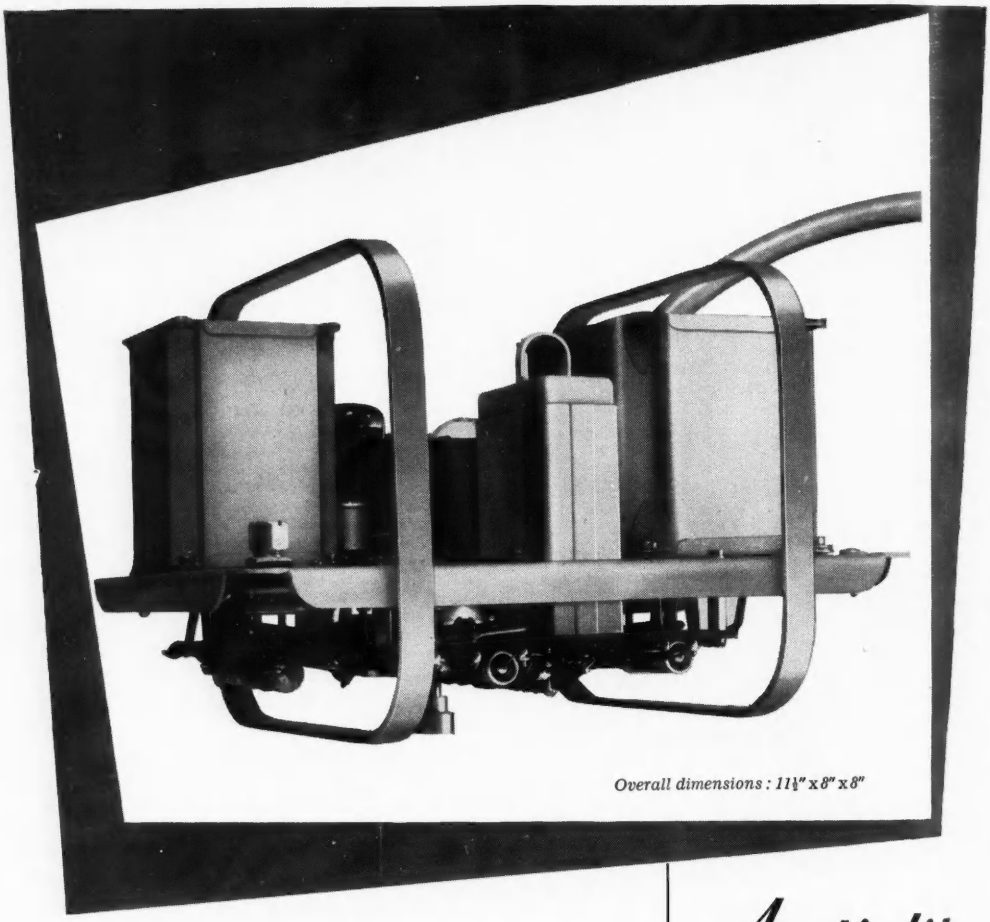
In order to make possible coolant pressures up to 1000 psi, these machines are equipped with a completely co-ordinated system consisting of a rear-spindle coolant inductor, high-pressure pump and gauge, and filters. The coolant shuts off automatically on the return stroke and can be switched off, together with the spindle motor, to permit accessibility for set-up. The chip-disposal trough is located immediately behind the bushing, and is 12 in. wide, to facilitate disposal of bulky chips. The main mounting of the machine is a cast-iron base with a trough, 3 in. in width, completely surrounding the unit and fixture areas.

MAGNETIC COOLANT SEPARATOR

A new magnetic coolant separator, recently added to their range of equipment by Rapid Magnetic Machines Limited, of Birmingham, has been specially designed to remove impurities, sludge, etc. from suds and other coolants.

The contaminated liquid is fed through a restricted

Synchronising AUTOMATICALLY



Overall dimensions : 11½" x 8" x 8"

The new Austinlite Synchronising Unit cuts out the need for complex and time-taking precautions when switching an A.C. generator into an already running A.C. system. The Unit detects voltages, phases and frequencies and when these are all in step with the system the generator is paralleled and can be safely switched in. More than this, with auxiliary apparatus additional generators can be automatically started up, synchronised, and connected with the system—and later disconnected—as the power demand fluctuates. The Austinlite Synchroniser can again be combined with other apparatus to form a load sharing device between two paralleled generators.
Simple in design, the Austinlite Synchroniser is self-contained, built on a small chassis suitable for back-of-panel mounting. Installation is an easy matter; little or no maintenance is required. It can, if required, be used as a conversion unit with existing generating plant.
Let us give you a more detailed description of this unit.

Austinlite

ELECTRICAL EQUIPMENT

Tailor made by
STONE-CHANCE LTD.

(makers of Sumo Pumps and Stone-Chance Lighthouses),
CRAWLEY, SUSSEX.

area between the separator housing and the stainless-steel cover of the magnetic drum. This drum, which is energized by powerful permanent magnets fitted around its periphery, rotates in the opposite direction to the flow of liquid. Ferrous particles in the liquid are thus attracted to the cover of the drum, and are discharged onto a chute at the front of the separator, after passing under a roller which squeezes out the liquid remaining in the sludge. The cleaned liquid flows from side outlets located near the front of the separator, and is then ready to be re-circulated. Magnetic materials adhering to the drum face form a mechanical filter, which also collects non-magnetic abrasive particles. For ease of cleaning, plugs are suitably located at points where sludge may collect.

This equipment, which is available in a range of capacities from 10 to 40 gpm, can also be adapted for extracting ferrous impurities from other liquids, including lubricating oils, potters' slip, chemicals, and paints.

PORTABLE TRANSISTORIZED METAL DETECTOR

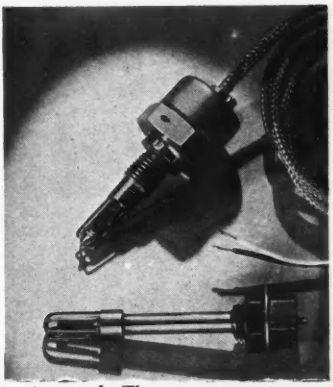
Complementing their range of heavy-duty metal detectors, Metal Detection Limited, of Birmingham, are now offering a portable lightweight detector incorporating transistors. Basically, the equipment comprises a hand unit and earphones, the unit (only 4 lb in weight) being passed over the surface or object suspected of containing metal and causing the oscillator volume to increase when any metal is directly beneath the hand-unit position.

This apparatus is very simple to operate and can be used for detecting metal-reinforcing in concrete, metal in timber, hidden metal roof-supports in mines, and many other applications in which it is desirable to locate hidden metals easily and reliably.

HIGH-SPEED PLATINUM RESISTANCE OPEN-TEMPERATURE PICK-UPS

A new line of high-speed precision platinum resistance thermometers, especially developed for temperature sensing in non-corrosive liquids and gases, is announced by Trans-Sonics, Inc., of Burlington, Mass. Because the environment to be measured is in almost immediate contact with the exposed platinum sensing element, extremely fast thermal responses of less than 2·5 sec in air and less than 50 millisec in water are obtained. These pick-ups are designed to operate over a temperature range from — 320 to + 750°F with an accuracy of better than ± 1% of full scale and a repeatability of 0·2%. In addition, they can deliver up to 5 V directly to telemetry commutation circuits without additional amplification. The pick-ups are designed to withstand pressures of up to 1500 psi ; resistance change is 100 ohms over the specified temperature range, and the maximum continuous current rating is 20 mA r.m.s.

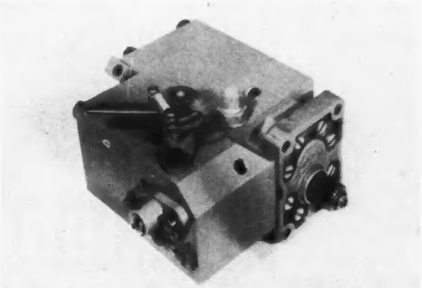
The platinum resistance winding, which is supported on mica cards or ceramic-insulated posts, is completely protected by a stainless-steel cage, and the body of the instrument, including the mounting thread and nut, is also of stainless steel. The mounting nut is machined to seat against a metal O-ring, to provide a pressure-tight seal in the vessel wall. Each instrument is furnished with teflon-covered leads protected with stainless-steel braided shielding, a special design feature allowing these leads to be removed, so that the external leads may be secured to the instrument terminals. In addition, leads may be soft-soldered or silver-soldered to the terminals.



FUEL-CONTROL UNIT FOR SMALL GAS TURBINES

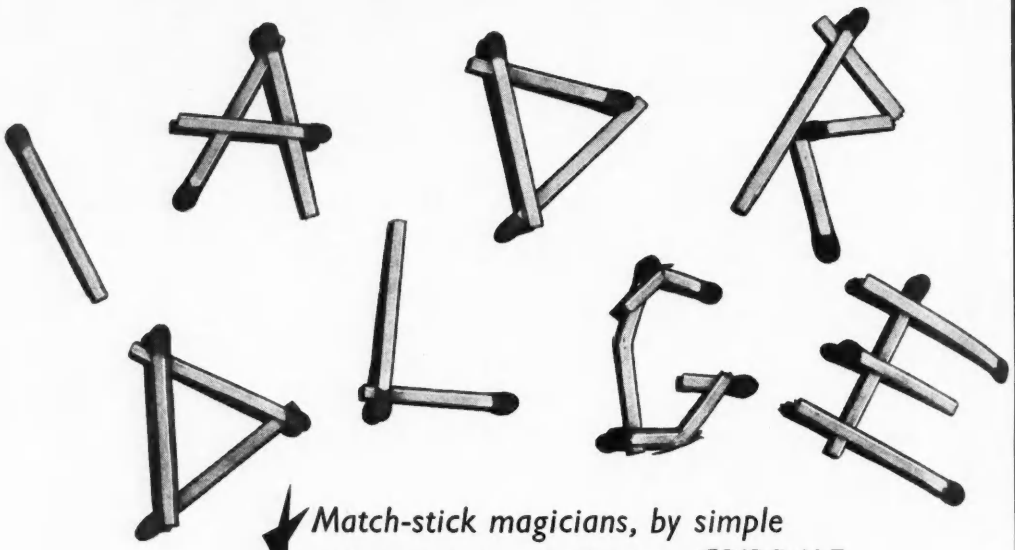
A completely self-contained fuel-control system for use on small gas turbines is under development by The Plessey Company Limited, of Ilford, Essex. Weighing 8 lb, the new system comprises a fuel pump, a governor by-pass valve, pressurizing and relief valves, and a shut-off cock, all built into one compact unit. Furthermore, pipework is cut to a minimum, comprising only the inlet from the fuel tank and delivery connections to the burners and igniter.

The unit is suitable for use on engines of 200 to 1000 bhp, with drive speeds between 3000 and 6000 rpm and fuel pressures up to 800 psi. It can be built to give any desired governor characteristic to suit the engine and its application. Accelerations are controlled by metering fuel proportionally to engine speed at the required rate. Maximum fuel pressure is controlled by a relief valve and, to prevent flame extinction following a sudden removal of load, a valve, to prevent pressure falling below the necessary minimum value, is also provided. The governor unit is generously proportioned, to ensure accuracy, while the fuel-control valves have no sliding parts, thus eliminating the possibility of hysteresis due to friction.



Speed setting is externally adjustable, and a bleed from the servo-chamber can be provided to allow the use of an overriding control, such as a Bourdon-tube-operated valve to control maximum temperature, or a solenoid-operated valve to control speed and therefore frequency, when the engine is used to drive an alternator.

Do you know the name for Plastics Mouldings?



Match-stick magicians, by simple manœuvre, can conjure up GRIDDALE or even DIRE GLAD, but knowledgeable Buyers know that they could not improve upon ALDRIDGE—*the name for plastics mouldings.*

ALDRIDGE is a comparatively new name in the moulded plastics field. It is a vigorous Company enjoying the benefits of the most modern factory and plant, and administered and staffed by men whose knowledge and experience of plastics mouldings extends back to the beginning of the Industry itself. ALDRIDGE make all types of thermo-setting and thermo-plastic mouldings for the leading names in many Industries . . . . . ALDRIDGE is ready to serve YOU!

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NEWS OF THE MONTH

PERSONAL

Mr. H. A. Luss has been appointed Chairman of the Nucleonics Group of the Scientific Instrument Manufacturers' Association. This appointment took effect as from the 6th July. Mr. Luss, who is at present Joint Managing Director of Isotope Developments Limited, London, E.C.2, has also been appointed Deputy Chairman of the company.

Associated British Engineering Ltd., London, W.C.1, announce that **Mr. Alexander Belch, C.B.E.,** has been appointed to the boards of their subsidiary Companies, H. Widdop & Company Ltd., Keighley, and the Free Piston Engine Company Ltd., London. Mr. Belch retired from his official appointments with the central commercial organizations of the Shipbuilding Industry in January last.

Mr. John William Mills, O.B.E., M.I.Mech.E., for over twelve years a Director of Armstrong Whitworth (Metal Industries) Ltd., Gateshead upon Tyne, and Jarrow Metal Industries Ltd., has decided to relinquish his seat on the board, in order to reduce his business commitments a little.

After 55 years, **Mr. Frank H. Farrer** has resigned from active participation in the affairs of The Villiers Engineering Co. Ltd., Wolverhampton, and the directors have appointed him as the company's first president. With his retirement from the chairmanship have come changes in the Villiers directorate. The new chairman is **Mr. Leslie W. Farrer** and the deputy chairman, **Mr. H. Geoffrey Jones,** while **Mr. T. P. Walmsley** has been appointed to the board, which is completed by **Mr. G. Teasdale,** who succeeded **Mr. Christopher C. C. Bridge** as London director last year.

Foxboro-Yoxall Ltd., London, S.W.19, will shortly open their new factory at Redhill, Surrey. To ensure that full advantage is taken of the increased production facilities which will be available, the Company is now engaged in a reorganization and expansion of the staff. First appointments to be announced are : **Mr. B. W. Balls, B.Sc., M.I.Chem.E.,** to be Technical Sales Manager, and **Mr. A. H. Isaac, B.Sc., A.M.I.Chem.E.,** to be Manager of the Sales Engineering Department.

Tecalemit Limited, Plymouth, announce that **Mr. A. Muir, M.I.Mech.E., A.S.M.A.,** has been appointed Manager for the Scottish and Northern Ireland Region, the Regional Office being situated at 86 Cambridge Street, Glasgow, C.2.

Westinghouse Brake & Signal Co. Ltd., London, N.1, announce the following appointments to the board :—**Mr. G. W. H. Richardson** ; **Mr. T. J. Aldridge,** Manager, Signal & Colliery Division ; **Mr. N. G. Cadman,** Works Manager ; **Mr. J. W. G. Kershaw,** Manager, Brake Division; **Mr. L. E. Thompson,** Manager, Rectifier Division.

Mr. R. J. Dixon and **Mr. S. W. Perkins,** who are Directors of Wickman Limited, Coventry, have now been appointed to the Board of Arthur Scrivener Limited, Tyburn Road, Birmingham 24.

Samuel Fox and Company Limited, a subsidiary of The United Steel Companies Limited, Sheffield, announce that **Mr. G. D. Illingworth** has been appointed Labour Manager from the 1st July. He succeeds **Mr. B. Butcher,** who was recently appointed a Director of Samuel Fox. In addition to his work as Labour Manager, Mr. Illingworth will be responsible for the engagement of labour, the safety and welfare departments and for works publications.

The following have resigned from the Board of The Brush Group Limited :—**Sir Ronald W. Matthews, D.L., M.Inst.T., Rt. Hon. Lord Cunliffe, Capt. R. C. Petter, M.I.Mech.E., M.I.Mar.E., Major Noel E. Webster, O.B.E., M.C., M.I.M.E., Mr. Miles Beevor.** The following have now been appointed to the Board of The Brush Group Limited :—**Sir Frank Spriggs, K.B.E., Hon. F.R.Ae.S., Sir Roy Dobson, C.B.E., Hon. F.R.Ae.S., J.P., John F. Robertson, C.A., Sir Arnold Hall, F.R.S., M.A., Hon. A.C.G.I., F.R.Ae.S.**

Edgar Allen & Co. Limited, Sheffield, regret to report the death of **Mr. Alfred Rubin,** of Zürich, who represented the firm for the sale of Trackwork in Switzerland for over 40 years, until his retirement in 1954.

Edgar Allen & Co. Limited have pleasure in announcing the appointment as Senior Representative in South and West Yorkshire, North Lincolnshire, etc., from May 1st, of **Mr. J. T. Wells, M.Eng., A.M.I.Mech.E.**

Ransome & Marles Bearing Co. Ltd., Newark-on-Trent, have been represented in Scotland for the past thirty-five years by **Mr. J. T. More.** At his own wish he is retiring at the end of this month. From the 1st July, Ransome and Marles' Area Manager will be **Mr. F. W. Austin,** with **Mr. E. H. Ward** as his assistant. Mr. Austin was formerly Branch Manager at Dundee. These representatives will be based at the new offices at 380 Argyle Street, Glasgow, where extensive bearing stocks are held. Stocks will continue to be maintained at 310 Perth Road, Dundee, under the supervision of **Mr. W. L. Driscoll.**

Mr. Frederick Cooper, A.M.I.Mech.E., has been appointed Chief Draughtsman of F. Perkins Ltd., Peterborough. Formerly Chief Draughtsman with Tecalemit Ltd., Plymouth, he was previously 15 years with the Napier group. His new duties will include responsibility for drawing office routine, final issue of production drawings and liaison with outside departments.

A new appointment—that of technical engineering adviser (Retaining Ring Division)—is announced by Geo. Salter & Co. Ltd., of West Bromwich. The post will be filled by **Mr. Martin M. Polson, B.Sc.,** whose duties will take him throughout the country advising firms on the possibilities of cost reduction by using retaining rings instead of traditional methods of fastening.

Mr. F. Davenport, assistant to **Mr. Harry Willshaw, O.B.E.,** Chief Engineer of the Dunlop Rubber Company, has been appointed Deputy Chief Engineer at their head office in London.

The Rt. Hon. Lord Moynihan, O.B.E., T.D., has accepted an invitation to join the board of directors of Simms Motor Units Ltd. as from the 28th May, 1957. Lord Moynihan is a Director of Durham Stokes R.B. Priest & Co., Stockbrokers, and also of Industrial and Commercial Holdings Group Ltd., who look after Simms' interests in South Africa.

NOW THERE ARE EIGHT

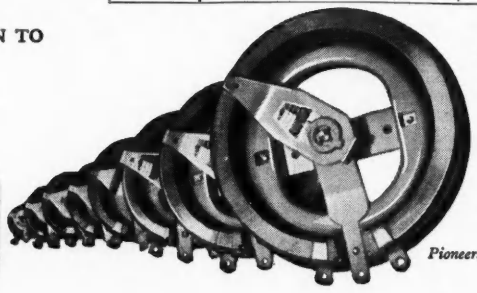
BERCO

TOROIDAL RHEOSTATS

So popular has the original range of five Berco rheostats become that three new models have been introduced. Like all Bercostats they have five important features:—

- VITREOUS ENAMEL BONDED
- HIGHEST MECHANICAL STRENGTH
- MAXIMUM ELECTRICAL PERFORMANCE
- MINIMUM SIZE IN RELATION TO POWER DISSIPATION
- CAN BE USED IN TROPICAL SITUATIONS WITHOUT MODIFICATION

Ohms ± 10%	CURRENT RATINGS							
	L.25	L.50	L.75	L.100	L.150	L.225	L.300	L.500
1	—	7.07	8.66	10.00	12.50	15.0	17.3	22.3
2	3.54	5.00	6.12	7.07	8.66	10.6	12.2	15.8
3	2.89	4.08	5.00	5.75	7.07	8.66	10.0	12.9
5	2.24	3.16	3.88	4.77	5.48	6.71	7.75	10.0
7.5	1.82	2.58	3.16	3.65	4.47	5.48	6.32	8.17
10	1.58	2.24	2.74	3.16	3.88	4.74	5.48	7.07
15	1.29	1.83	2.24	2.58	3.16	3.87	4.47	5.78
25	1.00	1.41	1.73	2.00	2.45	3.00	3.46	4.47
50	.707	1.00	1.23	1.41	1.73	2.12	2.45	3.16
75	.578	.818	1.00	1.15	1.41	1.73	2.00	2.58
100	.500	.707	.866	1.00	1.23	1.50	1.73	2.23
150	.408	.575	.707	.818	1.00	1.22	1.41	1.82
200	.354	.500	.612	.707	.866	1.06	1.22	1.58
350	.268	.378	.463	.535	.655	.803	.926	1.19
500	.224	.316	.388	.447	.548	.671	.775	1.00
1,000	.158	.224	.274	.316	.388	.474	.548	0.708
1,500	.129	.183	.224	.258	.316	.387	.447	.578
2,500	.100	.141	.173	.200	.245	.300	.346	.448
5,000	—	.100	.123	.141	.173	.213	.245	.315
7,500	—	.081	.100	.115	.141	.173	.200	.258
10,000	—	—	—	.110	.125	—	.173	.224
15,000	—	—	—	—	—	—	—	.163



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Sir W. H. Bailey and Co. Ltd., Patricroft, Manchester, have announced that they have appointed **Mr. R. A. Bartley, A.I.Mar.E.,** as Technical Sales Representative for the North Eastern area, including Northumberland, Durham, Cumberland, and the North Riding of Yorkshire. Enquiries relating to the company's wide range of sluice valves, reducing valves, test pumps and turnstiles should be addressed to Mr. Bartley, at 79 Ravensbourne Avenue, East Boldon, Co. Durham.

The directors of the Drayton Regulator and Instrument Company Limited announce that **Mr. P. H. Lloyd** has been elected Chairman of the Company, to take the place of the late **Mr. B. T. Wingfield** and that **Mr. R. E. Ashton** has been appointed Managing Director. **Mrs. N. L. Wingfield** and **Mr. A. M. Bell** have been appointed Directors.

Group Capt. John Alexander Kent, D.F.C. and Bar, A.F.C., A.F.R.Ae.S., M.I.Ae.S., 42, has been appointed Sales Manager, Kelvin & Hughes (Aviation) Ltd. This is a new post in the company.

Modern Wheel Limited announce the appointment as Technical Manager of **Mr. Ian Wans, M.I.Mech.E., M.I.Mar.E., A.M.I.C.E.** Mr. Wans, formerly of Ruston & Hornsby Limited, recently resigned from the Atomic Power Department of the English Electric Company Limited.

Mr. Stanley H. Hughes has been appointed Sales Manager of the Electrical Measuring Division of Elliott Brothers (London) Limited, Century Works, Lewisham. Mr. Hughes was formerly with Measuring Instruments (Pullin) Limited.

Mr. H. A. Lewis, M.B.E., T.D., B.Sc.(Eng.), A.C.G.I., M.I.E.E., A.M.I.I.A., has been appointed Managing Director of E.M.I. Sales & Service Ltd., Hayes, Middx., in succession to the late Mr. E. J. Emery. As from 1st June next, Mr. Lewis will in addition operate as Commercial Director of E.M.I. Electronics Ltd. Mr. Lewis became Personal Assistant to Mr. Emery in July, 1956, and was appointed a Director of E.M.I. Sales & Service Ltd. last December.

Mr. Gordon Richardson has been appointed Director of Head Wrightson & Company Limited, London, S.W.1.

Remington Rand Limited, London, W.C.1, announce the appointment of **Mr. Eric Segar** as Area Sales Supervisor (Supplies Division) for the entire London Region, as from 1st May, 1957. Mr. Segar joined the Company in London in October, 1950, as Supplies Sales Representative.

Mr. David W. Payn, T.D., F.C.C.S., has been appointed General Manager and Secretary to the Lead Development Association as from 20th May, 1957.

Newall Group Sales Ltd., Peterborough, announce the appointment of **Mr. Edwin C. Murphy** as Sales Controller of their Keighley Division. The appointment is consequent to the continued expansion of Keighley Grinders (MT) Ltd. and de-centralization from Peterborough of the sales organization of the Newall Group.

Air Control Installations Ltd., Ruislip, Middx., announce that **Mr. G. E. Clifford, B.Sc., Ch.E., A.I.Ch.E.,** has been appointed Technical Director, and that **Mr. H. C. S. Brand, M.A., A.I.I.A.,** has been appointed Works Director.

Desoutter Bros. Ltd., London, N.W.9, announce that **Mr. S. E. Wheadon, M.B.E.,** who was formerly Chief Technical Representative with the company, has been appointed Technical Sales Manager.

On Friday, the 31st May, **Mr. Ralph J. Cordiner,** President of the General Electric Co. of America, visited the Rugby Works of British Thomson-Houston and, accompanied by **Mr. E. H. Ball,** Managing Director, B.T.H. and **Mr. H. E. Cox,** Manager, Rugby Works, toured several of the factories.

Black & Decker Limited, of Harmondsworth, Middlesex, the manufacturers of portable electric tools, today announce the reconstitution of their board of Directors. **Sir Walter Puckey,** the well-known industrialist, joins the Board. **Mr. F. W. McCartney,** Assistant General Manager, becomes Assistant Managing Director, and **Mr. W. O. Bell,** Acting Production Director, becomes Production Director. **Mr. Robert Appleby,** the Managing Director, said the reconstruction of the Board was designed to give added strength to the company, and was in line with Black & Decker's future plans in the international field of operations.

Mr. Philip Bache, Managing Director of Geo. Salter & Co. Ltd., the West Bromwich spring and spring balance makers, has been elected President of Birmingham Chamber of Commerce for 1957–58.

Major J. Vivian Holman, A.F.R.Ae.S., M.I.Ae.S., F.Inst.D., has recently joined the Boards of Adam Engineering Company Ltd. (Small Tool Manufacturers), Vonarz Repetition Co. Ltd. (Precision Engineers, and Haesler Sales Company. His address will be 4 Grange Street, St. Albans, Herts. Telephone St. Albans 54336/7. Major Holman has also joined the Board of V.H.M.S. Enterprises Ltd. (Research and Development Engineers) of High Wycombe, Bucks.

British Oxygen Chemicals Limited, London, S.W.1, announce the appointment of **Mr. A. J. Oxley, M.Sc. (Sheff)** as Production Manager at their works at Chester-le-Street.

Mr. A. W. Morrison, B.Sc. (App. Chem.), D.R.T.C., A.I.I.A., has been appointed Director of Expandite Limited. Mr. Morrison joined the Company last year as Technical Manager in charge of Production and Development.

The Power-Gas Corporation of Stockton-on-Tees has just announced the appointment of **Major W. R. Brown, D.S.O.,** as Chairman and Managing Director in succession to the late Dr. N. E. Rambush whose death occurred on the 15th May, 1957. Major Brown has been associated with the Stockton company for 30 years, and was appointed to the board of directors in 1942 and was elected Vice-Chairman last year.

Mr. M. H. Hammill, B.Sc., A.M.I.Mech.E., who has been Director and General Manager of Foster Transformers Ltd., South Wimbledon, S.W.19, since 1950 has been appointed Managing Director. Before joining the Foster Company—which is one of the Lancashire Dynamo Group—he was associated for many years with the industrial consultants, Urwick Orr & Partners.

Martonair Limited are pleased to announce the appointment of **Mr. V. Chidgey** as Technical Representative for Wales.

Follsain-Wycliffe Foundries Ltd. announce that **Mr. W. A. Murray,** 147 Bath Street, Glasgow, C.2., telephone Central 5374, has been appointed Representative for Scotland. **Messrs. Barrett, Greene & Co. Ltd.,** Walton, Stone, Staffs. (telephone Stone 11 and 550) have been appointed as Representatives in North Wales, the Potteries, Shropshire and South Cheshire.

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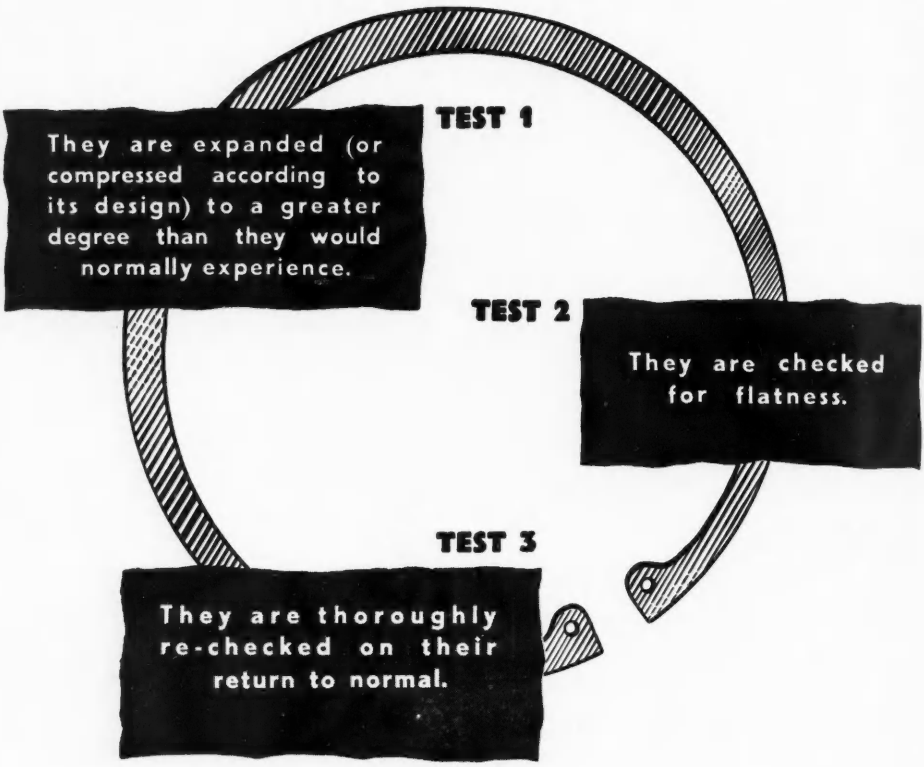
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BUSINESS NEWS

Aero Research Limited, Duxford, Cambridge, manufacturers of adhesives and honeycomb for aircraft structures, have produced a 16 mm colour sound film describing the " Redux " bonding process. The method of use of Redux and the characteristics of bonded structures are described in detail and typical bonded components and bonding plants are shown. Reference is also made to honeycomb sandwich structures. The film, which runs for 17 minutes, is available in French, German, Italian and Japanese (for magnetic reproduction), as well as in English (optical reproduction). Copies are available on loan to aircraft manufacturers, technical colleges and other interested organizations from Aero Research Limited.

The Departments of Civil and Mechanical Engineering of the University of Nottingham are holding a course on **" Nuclear Engineering "** from the 16th to 20th September, 1957. The object of the course is to present an outline of Nuclear Engineering to engineers with little or no previous knowledge of the subject. Information may be obtained from Mr. T. L. O'Neill, Secretary to the Course.

Henry Hargreaves & Sons Ltd., Bury, Lancs, announce the opening of a London Office as from the 24th June 1957, at 603 Grand Buildings, Trafalgar Square, W.C.2. Telephone : Whitehall 4805. Mr. Cyril F. Morgan, L.R.I.B.A., A.M.I.Mech.E., who is in charge of this office, has joined the Company as Technical Manager for the London Area.

The **Report on the Opening and Industrial Sessions** of the Joint Conference of the Combustion Engineering Association and the Solid Smokeless Fuels Federation on **" The Application of the Clean Air Act to Private Dwellings and Industry "** held at the Queens Hotel, Birmingham, on the 16th and 17th April, 1957, is now available on application to the Combustion Engineering Association, 6 Duke Street, St. James's, London, S.W.1. Price 15*s.* to non-members of the Association.

On the 1st June, 1957, the Directorate of Colonial Surveys was renamed the Directorate of Overseas Surveys ; and the Colonial Geological Surveys was renamed the Overseas Geological Surveys.

Durham Raw Materials Ltd., London, E.C.3, announce that from the 1st June they have been appointed the sole selling agents in this country for the products of **Goodrich-Gulf Chemicals Inc.,** U.S.A. This change is by mutual agreement between the producers and the existing agents, **British Geon Ltd.**

Honeywell-Brown Ltd. have moved their Manchester Branch Office to substantially larger premises at Warsall Road, Northenden, Manchester, 22. The improved facilities are available for all enquiries concerning industrial instrumentation, heating and air controls, and precision switches.

The Plessey Company Limited announce that production of their standard Communications Equipment for Industrial, Commercial, and Services applications, has been moved from the Ilford, Essex, factory to the St. Ives Unit, West End Mills, Ramsey Road, St. Ives, Huntingdonshire, telephone St. Ives 2095. Mr. P. A. Tremaine, who has been with the Company for some years, remains in charge of this operation as the Unit Manager and is based at the new address to which all future enquiries should be made.

Dendix Brushes Ltd., Chepstow, Mon., in co-operation with their American Associates, the **Osborn Manufacturing Company,** Cleveland, Ohio, have, over the past two years been carrying out intensive research into the problem of evolving a special wire for use in rotary brushes, and can now offer a full range of rotary brushes filled with crimped wire which provide the necessary reliable characteristics of cutting power, safety with speed of rotation, plus operating life far in excess of the standard wire brush. This new range of rotary brushes will be known as the Dendix-Osborn Supermatic range.

It is announced by the **Institution of Production Engineers** that a specialist Group on Materials Handling has been formed within the Institution, under the chairmanship of Mr. A. G. Hayek, M.I.Prod.E., of A. G. Hayek and Partners, Ltd., Stoke-on-Trent. The purpose of the Group is to provide a central forum for those members of the Institution having a special interest in Materials Handling.

The London offices and sales depot of **Sir W. H. Bailey and Co. Ltd.** have been removed from 4 Domingo Street, E.C.1, to new and improved premises erected at Selinas Lane, off Whalebone Lane South, Dagenham, Essex, telephone DOMinion 2277/8/9.

Pencol Pipeline Engineering Consultants, 19 Grosvenor Place, London, S.W.1, has been formed by a group of experienced engineers as independent consultants. The services of the Pencol Organization are available for the design and engineering of pipelines, pumping stations and ancillary equipment in co-operation with those undertaking the development of pipeline and pumping projects of all kinds.

On the 1st June the **Blackman Export Co. Ltd.,** a subsidiary of **Keith Blackman Ltd.,** manufacturers of fans, ancillary equipment and industrial gas apparatus, moved office from 23 Queen Square, London, W.C.1, to the parent Company's Head Office at Mill Mead Road, London, N.17. At the same time Keith Blackman Ltd. opened a London Area Sales Office at the Queen Square address vacated by the Blackman Export Co. Ltd.

After more than a quarter of a century at Walnut Tree Walk, S.E.11, **Charles Churchill & Co. Ltd.,** manufacturers and distributors of machine tools and engineers' supplies, moved on 1st June, to new premises in Edgware Road, The Hyde, N.W.9.

The new premises, of modern design, are located on the main thoroughfare into London from the North, close to the North Circular Road. They occupy some 28,000 square feet of floor space and provide commodious office accommodation and showrooms for the display of machine tools and small tools, in addition to greatly extended facilities for warehousing purposes.

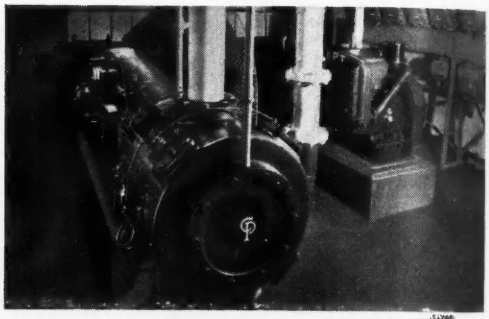
Clarkson (Engineers) Ltd., have opened a new Branch Office and Stockroom at Room 8A Britannia House, Leeds 1, telephone 26369. This will improve their coverage of Yorkshire, as full stocks of Clarkson Autolock and Dedlock Chucks and Milling Cutters, also the range of Clarkson Spiral Side and Face Cutters are held there, and telephone orders will be despatched the same day.

The growing demand for test equipment manufactured by **Marconi Instruments Ltd.** has made it necessary for the Company to provide a 22,000 square feet extension to the factory at Longacres, St. Albans, Herts. Besides considerably expanding their sales in the home market in recent years, the Company have made big advances in the export field and now have distributors in sixty-eight countries.

In order to keep pace with the growing demand for " Airflo " products including P.V.C. Fans, Hoods and Ducting, Ventilators, Welding Tables and Dust Collectors, **The Industrial Fan & Heater Co. Ltd.,** of Birmingham 11, a Member of the Simms group of Companies, has opened a new branch works at Dawley Brook, Stourbridge, Worcestershire.

Horizontal water cooled compressors running quietly and effortlessly virtually for ever

Complete reliability with little or no attention is what is required of a compressed air plant for permanent installation. This means a slow speed machine—such as the class T range made by Consolidated Pneumatic. Horizontal, water cooled machines, running quietly and effortlessly—virtually for ever at 275 r.p.m., they are built to give the service and performance demanded ; and with their finely controlled lubrication system, deliver the cleanest air. For constant, compressed air power with negligible maintenance needs, investigate class T Consolidated Pneumatic compressors. A big variety of capacities and pressures available in single and 2-stage models. Write for information on CP slow speed class T compressors.



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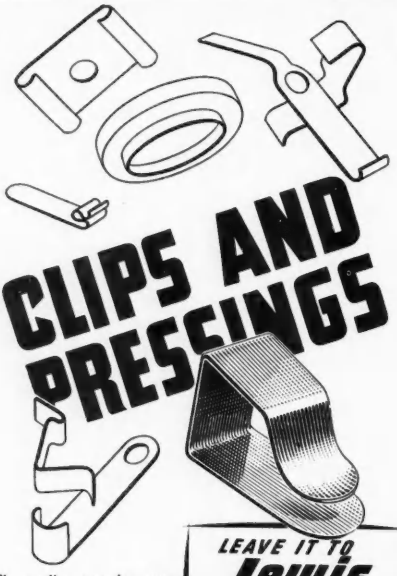

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NEW WIND TUNNELS FOR AIRCRAFT RESEARCH

Three wind tunnels are now in use at the Royal Aircraft Establishment near Bedford. The project, which started in 1947 with the construction of the 3 ft by 3 ft transonic and supersonic tunnel, is now complete following the recent final acceptance testing of the 8 ft by 8 ft High Speed tunnel.

TRANSONIC AND SUPERSONIC TUNNEL

The transonic and supersonic tunnel is designed to test models in its 3 feet square working section at air flow speeds ranging from subsonic to Mach 2. It comprises a steel pressure shell which forms a closed circuit around two centrifugal compressors. The necessity of attaining the desired aerodynamic performance made it essential to achieve a smooth surface finish and a greater degree of accuracy on cross-sectional dimensions that was then generally considered possible on pressure vessels of this size. To maintain the desired operating temperature in the working section, coolers were built into the circuit. The tunnel structure was designed to operate at pressures between near vacuum and 15 psig. Due attention was given to the dynamic loading caused by changes of direction of the air stream and to the indeterminate loading due to buffeting of the stream. The support system was designed to allow freedom of movement, roller and pinned members being used so that the numbers of expansion joints could be kept to a minimum. As the tunnel was isolated structurally from the spring mounted compressors, flexible joints were used to connect up. This isolation imposes horizontal loads on the structure, so that some supports had to be fixed to carry the loading into the foundations. The working section and adjacent diffuser were built on mobile trolleys as separate units and bolted into position on the circuit. Originally the working section was supplied with nine interchangeable top covers, each lined to give the throat a shape appropriate to a given air speed. Since the tunnel was put to work, further top covers and a complete interchangeable working section and balance section have been supplied to provide for transonic speeds and a range of supersonic speeds up to Mach 2.

VERTICAL SPINNING TUNNEL

The second installation is the vertical spinning tunnel which is used to test the behaviour of free flying models and to study methods of recovery from aircraft spin. It is a closed circuit tunnel of much different form from the 3 ft × 3 ft and 8 ft × 8 ft tunnels, insofar as the air in the working section travels vertically upwards and at speeds in a much lower range (about 80 mph) than those obtaining to the other tunnels. A fan at the upper end of the 15 ft diameter working section circulates the air, which returns to the lower end through a concentric diffuser outside the working section. The outer shell of this diffuser is the pressure container which allows the tunnel to be operated between atmospheric condition and 45 psig. The inner structure carries working and observation floors and a chamber from which personnel can observe and control the tunnel and model launching during tests at the higher pressures. The whole of this internal structure is carried from the roof of the pressure shell, which is reinforced with arch girders for this duty. The superstructure built over the top of the pressure shell houses the fan motor and control gear. This motor room is complete with overhead gantry for servicing the mechanical gear. The space between the motor room floor and the pressure shell dome has been conveniently used for normal water tower purposes and has a capacity of 50,000 gallons. Absolute verticality is important to the operation of the tunnel. Means have, therefore, been provided to correct uneven settling during its life by jacking the whole structure at the column footings.

HIGH SPEED TUNNEL

The high speed tunnel is intended to supply information for the development and evaluation of aircraft in both the subsonic and supersonic speed ranges ; these cover speeds from zero to Mach 2·7. This tunnel is essentially a pressure vessel generally of circular cross section varying in diameter throughout its length and forming a closed circuit with four rectangular corners. The total weight of steel is about 5000 tons and it is claimed that it is the largest pressure vessel in Europe. The structure is capable of bearing internal pressures ranging from complete vacuum to 45 psig at which pressure the energy stored in the shell is equivalent to that needed to lift 5000 tons about 350 ft. The air is driven round the tunnel circuit by an axial flow compressor ; the 80,000 horse power expended in this machine heats the large volume of air which passes through it and this heat must be removed before the air passes to the working section in which the models are tested. For this purpose a large cooler was built into the tunnel circuit which at this point is 47 feet diameter. The cooler contains 58,000 separate tubes, the total length of which is 180 miles. Thermal movements of the various tunnel sections between the fixed points at compressor and cooler necessitated the use of five expansion joints in the shell. Some, 20 ft in diameter, were on the bellows principle ; others, where the movement amounted to about 5 in., were of the sleeve type with inflated rubber seals. The sleeve type expansion joints required accurate machining and assembly of heavy rings over 16 ft in diameter, together with mechanical gear for manipulation during subsequent service to the rubber seals. Supports for the structure were designed so that movement could take place where required, while at points where full fixity was required horizontal thrusts of about 1200 tons were carried through to the foundations. Much exacting work was carried out on internal fairings to produce the required air flow at points such as the exit from compressor, the rectangular corners and the entry to the working sections after the cooler. In these fairings a high degree of accuracy was required on ordinates and slopes together with smooth contours free from waves or irregularities at the welded joints. As with all tunnels of this type, only the subsonic flow, that is that below the speed of sound, Mach 1, can be controlled by changing the compressor speed. Supersonic flow can only be developed and regulated by the introduction of a suitably shaped nozzle into the air flow at the " working section " where the model is under test. In all previous tunnels in this country the nozzle form has been achieved by the use of formed blocks, a different block being necessary for each speed. The main feature of this tunnel however is that the required nozzle shape is created by the flexing of 1 in. thick steel plates, 62 ft long by 8 ft wide, which form the roof and floor of the working section.

Among the contractors for the wind tunnels are Whessoe Ltd., London, S.W.1, who were responsible for the structural design, fabrication and erection, and Costain-John Brown Ltd., who designed and installed the flexible wall " throat " used to develop and regulate the supersonic wind speeds. They were also responsible for the supersonic diffuser designed to slow the airstream down to below the speed of sound. The electrical system for this supervisory control of the air speeds was designed and built by The General Electric Co. Ltd., while a subsidiary Company, The Reliance Telephone Co. Ltd., was responsible for the installation of the apparatus racks, the control desk and the apparatus within the tunnel, together with a large proportion of the cabling for the instrumentation.

Multispindle Drilling Machine

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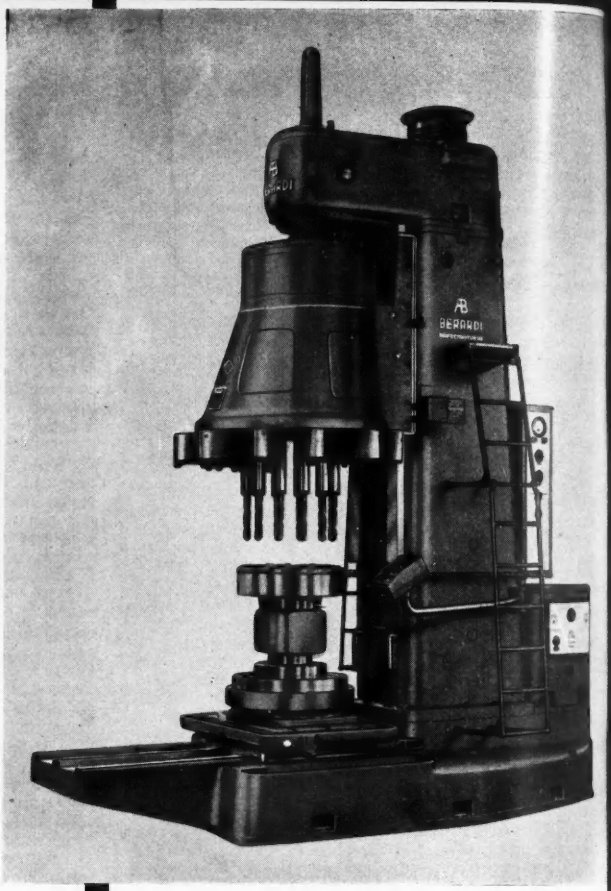
provided with 16 adjustable spindles on brackets, each of 40 mm diameter capacity. Hydraulic operated head working on automatic cycle. Each spindle has individual change gear box. Table is of the sliding type, to make easy loading and un-loading.



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Spindle drives n. 40, drilling capacity up to	23 mm. ea.
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or n. 16 drilling capacity up to	40 mm. ea.
Max. drilling range	1000 × 550 mm.
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Head travel	mm. 600
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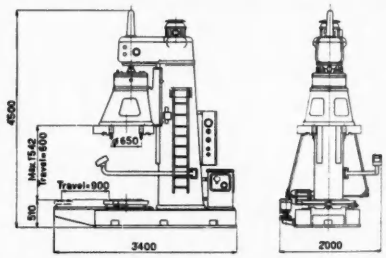

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CONTRACTS

Orders to a total value in excess of £100,000 have been placed by **Polish Official Purchasing Agencies** with the **United Construction Machinery Co. Ltd.**, West Drayton, Middx., which had the biggest exhibition stand in the British section of the recent Poznan Fair. The equipment ordered includes Mono-Rail transporters, dumpers, excavators, weighbatchers, bulldozers, hoists, loading shovels, trailers and mobile cranes. All deliveries will be completed by January, 1958. In the intervening months there will be an interchange of visits between Polish technicians and experts of the U.C.M.C. companies. At the conclusion of the Fair, U.C.M.C. sold most of the exhibition equipment taken to Poland, to a value of about £50,000.

Work will start at the end of this year on a £135,000 Tower Box Purifier to be installed at the Worcester Gas Works for the **West Midlands Gas Board. Henry Balfour & Co. Ltd.**, Leven, Fife, have received a contract to erect this plant. Initially the plant will have a total oxide capacity of 28,400 cu ft, and the gas throughput will be 4,000,000 cu ft per day. Eventually it is hoped to double this throughput by adding six further oxide-holding trays to the east tower. The installation is scheduled to be completed by the end of 1958. Henry Balfour have already installed Tower Box Purifiers at the Southall, Bow Common, Rotherham and Beckton Gas Works.

The contract for the heating and cooling system of the new synthetic resin plant at **Styrene Co-Polymers Ltd.** has been awarded to **Hygrotherm Engineering Ltd.** The heating and cooling plant will consist of a Hygrotherm heat generator with a ring main and loop circuit system. Pneumatically operated three-way valves will control the temperature at which the heat transfer medium is supplied to the reaction vessels of the resin plant. The heating medium will be Shell Voluta 45. An outstanding feature of the heating and cooling systems will be the simplicity and precision of the temperature control which will be within ± 1°C.

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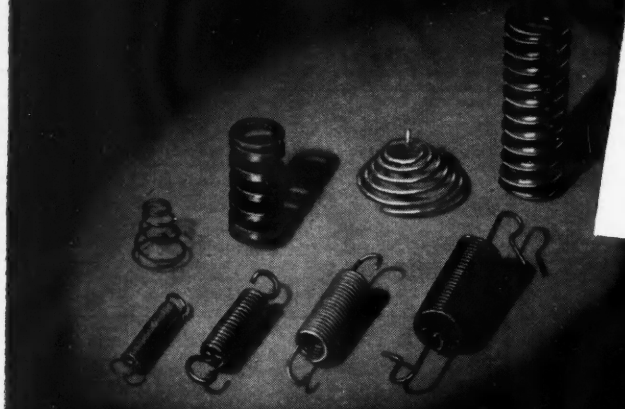

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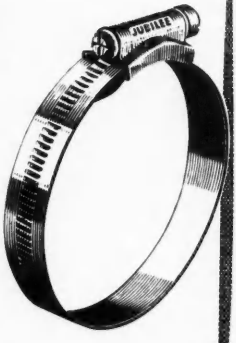

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BOOKS RECEIVED

Antennas. Edited by Alexander Schure, Ph.D., Ed.D., 88 pp., 55 illustrations. Publishers : John F. Rider Publisher, Inc., 116 West 14th Street, New York 11, N.Y., U.S.A. Price : $1.50.

A knowledge of antenna theory and fundamental antenna principles is obviously important to the serious student and practitioner of electronics, as an antenna constitutes the final link in a transmitting facility and the initial link in the receiver. Serious deficiencies may arise in a communications system, owing to improper choice of an antenna or lack of understanding as to the design, installation, matching requirements, and properties of an antenna furnished or recommended for use with a given electronics system.

The purpose of this book, therefore, is to provide the fundamental concepts of antenna theory. A minimum of mathematical treatment has been employed, but the analyses are sufficiently extensive to permit the interested technician, practising engineer, or advanced student to develop a full comprehension of the pertinent facets of such theory. To ensure this aim, specific attention is given to the fundamental antenna principles ; the basic antenna types ; input impedance and radiation resistance ; ground effects ; variations of electrical length by loading ; gain and directivity ; driven and parasitic arrays ; various types of long wire antennas ; feeding and matching principles ; and several variations on the basic dipole design reflected in current practical antenna types.

Wave Propagation. Edited by Alexander Schure, Ph.D., Ed.D. 64 pp., 23 illustrations. Publishers : John F. Rider Publisher, Inc., 116 West 14th Street, New York 11, N.Y., U.S.A. Price : $1.25.

A thorough understanding of the basic principles of wave propagation is a most useful tool for the electronics technician, engineer, or amateur. A working knowledge of essential wave-propagation factors is not too difficult to attain, and this book is intended to accomplish this aim by helping the student to understand the more important ideas pertaining to wave propagation.

Specific attention has been given to the nature and use of the electromagnetic wave in such areas as wave motion, frequency-wave-length relationships, radiation, field intensity, wave fronts, and polarization. Analyses are given of the natures of sky waves, ground waves, wave reflection, refraction, and diffraction as they pertain to the propagation of electromagnetic waves. A general discussion of the effect of atmosphere on radio transmission includes details of the regions which exert different influences on the passage of an electromagnetic wave, i.e., the ionosphere, stratosphere, and troposphere. In addition to an explanation of ground-wave and sky-wave propagation, some details are given dealing with scatter propagation. In short, the total materials presented are sufficient to give the essential elements of wave-propagation theory.

LATEST INDUSTRIAL LITERATURE

1. Contour Plotting Machine. This 8-page illustrated booklet describes a mechanical contour plotting machine which enables highly accurate true-to-scale drawings of any irregular shape to be plotted in a third of the time needed for hand drawing. The machine plots from true size ordinates with the accuracy of ± 0·002 in. at 20 times full size. The resulting drawing is within 0·0001 in. of the true size and is thus particularly suitable as a master drawing for inspection use by comparison with an optical projection of any contour or part. The machine is screwed to the top of a standard " Antiquarian " size drawing board and has two carriages that move over tubular guides to cover a 30 in. by 20 in. drawing area.

2. Alloy Tool and Die Steels. The field of usefulness of carbon tool steels, though considerable, is limited by the absence of special properties required in tools employed for particular purposes. By adding alloying elements such as manganese, nickel, tungsten, vanadium and molybdenum, in controlled amounts and combinations, a variety of tool steels may be obtained suited to the wider field of cutting, forming, battering, chipping and hot working operations for which carbon steels are unsuitable. This 38-page catalogue, describing a range of alloy tool and die steels, covers such factors as forging range, annealing, hardening, tempering and typical applications.

3. Hydraulic Hand Pump. Designed as a robust, hand-operated, pump to power all rams and attachments, this industrial equipment is illustrated in an 8-page brochure. Three pumps are available ; the standard size model will operate into a load capacity of 4, 7, 10, and 20 short-tons ; a slightly larger version will handle a load capacity of 50 short-tons while a " Bantam " model, of 4 short-tons capacity, is also available. A high-pressure hose transmits the hydraulic oil from the pump to the ram attachment, the length of the hose enabling inaccessible places to be reached.

4. Magslips—Applications and Methods of Use. This is a 60-page technical handbook, well written and profusely illustrated covering magslips and synchros. A brief introductory note on the history of remote indicating devices leads into the section dealing with the development and theory of operation of the magslip. Among the varied aspects of magslip usage covered in this interesting volume are :—magslip systems and circuits ; remote control and indication ; the design and layout of new applications ; measurements, tests and fault location. The book concludes with a comprehensive subject index.

5. Precision Pins. This very detailed 6-page leaflet describes an extensive range of pins, rivets, axles and special cut wires that the manufacturers have produced to special order. The purpose of the leaflet is to illustrate the specialized service provided in this particular field of engineering. Facilities for deburring, polishing and electro-plating are also available.

6. Metal Sawing Machines. The outstanding features of this hydraulic resistance metal sawing machine are that the majority of materials, irrespective of size, can be cut with blades of one pitch of tooth, that the cutting pressure is instantly regulated by the control handle and that the pump action on the return stroke lifts the blade well clear of the face of the cut. Six models are described in this 4-page leaflet, and cover a capacity from 6 in. by 6 in. to 20 in. by 8 in . in square and other sections.

7. Glass Fibre Reinforced Plastic Fabrications. This well illustrated brochure describes the use of furane laminates in the construction of tanks for almost all acids, alkalis, inorganic salts and organic solvents with the exception of oxydizing agents. This system is suitable for use up to a working temperature of 100°C. An outstanding feature of the design of the products is the " twin-shell " method of construction where a furane resin is used on the inside and a polyester resin on the outside. As well as for tanks the " twin-shell " system is ideal for the manufacture of ducts for corrosive fluids and chemicals.

8. Ventilation. Information on a wide range of ventilators is contained in this most informative 30-page catalogue. The three sections cover ventilators for constant heat and fume conditions, round ventilators for natural ventilation and fan types for accelerated ventilation. Several pages of information cover the design and application problems, while various tables enable the most suitable models to be selected, taking into consideration such factors as wind velocity, temperature difference, weight, noise levels, etc. A useful table indicates the recommended air changes for typical buildings.

9. Air Filter Units. This 4-page leaflet describes a new range of air filter units using glass fibres for filtration purposes. Designed to give the highest possible surface area of fibre combined with low binder content, they incorporate a new viscous fluid adhesive. The glass composition used is highly resistant to most chemicals and the filter can be used for the normal range of air conditioning temperatures ; optimum results are obtained between 0°F and 150°F. The normal dust holding capacity is of the order of 3 to 4 months. For high efficiency applications the 2 in. units can be arranged in tandem, operating at a velocity of 300 ft/minute, when an efficiency of 98% can be expected at 3·5 to 7 micron particle size. Tropically treated units are available.

10. Multiple Nutrunners. A new method of overcoming the design and construction problems in manufacturing a multiple nutrunner with a torque greater than 50 lb/ft is described in a 12-page illustrated catalogue. The problem which demanded a larger motor and more air has now been obviated by using a small motor having a low air consumption, and coupling this to an automatic two-speed gear box. The motor runs the nut down in high gear and, at a predetermined low torque setting, the gearbox automatically changes into a very low gear. This enables the motor to complete the final tightening to a high torque value. Torque control is effected by adjustment of the air pressure. The units are designed and made individually to suit each particular requirement, and may incorporate a number of refinements such as push-off plungers, etc.

11. V-Belt Maintenance. This new 20-page V-Belt maintenance manual will be of special interest to all those interested in the design and servicing of V-belts. This manual deals with the factors governing V-belt design drive and shows how, by observing a few simple precautions, V-belt users can prolong the life and increase the efficiency not only of the belts themselves but of the driving and the driven machinery. It is pointed out how correct installation and maintenance can confer benefits out of all proportion to the effort involved ; maximum life is obtained from every belt, stoppages and breakdowns reduced with a consequent saving in labour and machine time, production increased and production costs reduced. After dealing with the constructional methods the manual deals with the installation and care of belts, discussing such basic factors as alignment, adjustment, tension, storage, fitting and cleaning. A section is devoted to the operation of V-belting in hot, oily, corrosive or explosive atmospheres.

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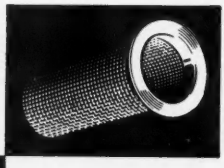

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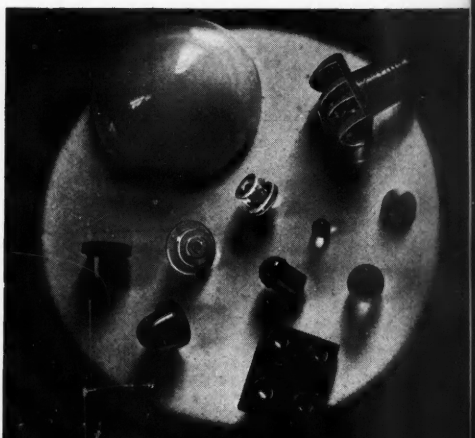

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